TM 5-858-4

TECHNICAL MANUAL

DESIGNING FACILITIES TO RESIST NUCLEAR WEAPON EFFECTS

SHOCK ISOLATION SYSTEMS

HEADQUARTERS, DEPARTMENT OF THE ARMY
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***This manual together with TM 5-858-1; TM 5-858-2; TM 5-858-3; TM 5-858-5; TM 5-858-6; TM 5-858-7; and TM 5-858-8 supersedes TM 5-858-1, 1 July 1959; TM 5-858-2, 15 March 1957; TM 5-856-3, 15 March 1957; TM 5-856-4, 15 March 1957; TM 5-856-5, 15 January 1958; TM 5-856-6, 15 January 1960; TM 5-856-7, 15 January 1958; TM 5-856-8, 15 January 1960; and TM 5-856-9, 15 January 1960.**

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CHAPTER 1

INTRODUCTION

1-1. General.

a. This series of manuals, entitled Designing Facilities to Resist Nuclear Weapon Effects, is organized as follows:

TM 5-858-1	Facilities System Engineering
TM 5-858-2	Weapon Effects
TM 5-858-3	Structures
TM 5-858-4	Shock Isolation Systems
TM 5-858-5	Air Entrainment, Fasteners, Penetration Protection, Hydraulic-Surge Protective Devices, EMP Protective Devices
TM 5-858-6	Hardness Verification
TM 5-858-7	Facility Support Systems
TM 5-858-8	Illustrative Examples

A list of references pertinent to each manual is placed in an appendix. Additional appendixes and bibliographies are used, as required, for documentation of supporting information. Pertinent bibliographic material is identified in the text with the author's name placed in parentheses. Such bibliographic material is not necessary for the use of this manual; the name and source of publications related to the subject of this manual is provided for information purposes.

b. The purpose of this series of manuals is to provide guidance to engineers engaged in designing facilities that are required to resist nuclear weapon effects. It has been written for systems, structural, mechanical, electrical, and test engineers possessing state-of-the-art expertise in their respective disciplines, but having little knowledge of nuclear weapon effects on facilities. While it is applicable as general design guidelines to all Corps of Engineers specialists who participate in designing permanent military facilities, it has been written and organized on the assumption a systems-engineering group will coordinate design of the facilities.

c. Technical Manual 5-858 addresses only the designing of hardened facilities; other techniques to achieve survival capacity against nuclear weapon attacks are deception, duplication, dispersion, nomadization, reconstitution, and active defense. A facility is said to be hardened if it has been designed to directly resist and mitigate the weapon effects. Most of the hardening requirements are allocated to the subsidiary facilities, which house, support, and protect the prime mission materiel/personnel (PMMP). This manual is applicable to permanent facilities, such as those associated with weapon systems, materiel stockpiles, command centers, manufacturing centers, and communications centers.

d. The nuclear weapon threats considered are listed below. Biological, chemical, and conventional weapon attacks are not considered.

—Weapons aimed at the facility itself or at nearby targets

—A range from many, relatively small-yield weapons to a single super-yield weapon

—Weapon yields from tens of kilotons to hundreds of megatons

—Weapon delivery by aerial bombing, air-to-surface missile, surface-to-surface missile, or sattelite-launched vehicle

—Detonation (burst) of a weapon in the air, at the ground surface, or beneath the ground surface

—Direct-overhead bursts for a deep-buried facility

—Near-miss bursts for a near-surface facility, producing peak over-pressures from tens to thousands of psi at the facility

e. The designing of facilities resistant to nuclear weapon effects is an evolving specialty using a relatively narrow data base that incorporates both random and systematic uncertainties. The range of these uncertainties may vary from significant (order of 1 to 2 magnitudes) to normal (10 to 100 percent variation from average values). The applicable uncertainty value depends on the specific weapon effect or hardening objective under consideration. Loading uncertainty is generally more significant than resistance uncertainty. Awareness of the appropriate uncertainty (extent of ignorance) factor is essential not only for system engineering trade-offs, but in the utilization of available analysis or test procedures. Studies and experiments are being conducted to improve methodology, to better define random uncertainties, and to reduce systematic uncertainties. This manual will be revised as significant improvements occur in either methodology or data base.

1-2. TM 5-858-4. Shock isolation systems.

This volume describes methods for designing shock isolation systems that mitigate the shock and vibration induced by a nuclear weapon attack. The volume includes a review of shock isolation system designs and presents shock and vibration tolerance data for personnel and generic equipment.

CHAPTER 2

SHOCK ISOLATION OBJECTIVES

2-1. Introduction.

a. Airblast and ground shock from a nuclear weapons attack can cause severe in-structure motion in hardened facilities. Although the structure itself may survive the attack, there may be unacceptable damage to equipment and injury to personnel. Shock isolation is needed to reduce the magnitude of motion transmitted to shock-sensitive equipment or personnel.

b. An overall objective of a shock isolation system for protective structures is therefore to mitigate the airblast and ground-shock effects induced by nuclear weapons. Aboveground structures are subjected to the direct effects of airblast as well as the movements of the ground beneath and around the structure. Underground structures are subjected only to the effects of ground shock. Cratering and debris impact are other relevant environmental effects. These requirements are summarized in table 2-1. See TM 5-858-1, table 3-1, for the complete list of facility objectives, including those for the structure and support systems.

c. To guide the shock isolation system design effort, an explicit statement of objectives is needed early in the design process. These objectives are developed and implemented in the following steps:

(1) Identify the functions, equipment, and personnel that are critical to mission success.

(2) Describe the facility environments at the locations of critical equipment/personnel.

(3) Determine the equipment and personnel tolerances to shock.

(4) Compare and evaluate the environments with respect to equipment/personnel tolerances.

(5) For those equipments/personnel needing protection, design a shock isolation system that fully meets the requirements and constraints imposed on the system.

These steps serve to limit the shock isolation design effort to functions that are important enough to need protection. Then, of the important functions, those not needing protection are identified and the necessary degree of protection for the remainder is indicated. This provides a sound basis for proceeding with design.

d. The remainder of this chapter will briefly discuss these five steps.

2-2. Identification of critical functions.

a. The nature of the prime mission will helo identify those functions critical to satisfactory mission performance. Typical prime-mission functions include reception and transmission of messages for command and control, weapon-system storage and launch, and data management and interpretation. Hardened facilities are also used for such non-prime-mission functions as the storage of equipment, the stockpiling of materiel, and the housing of personnel reserve troops.

b. Mission analyses will be conducted by the system engineering group during concept formulation and contract definition. These analyses will establish the basic functional and technical requirements that will guide system design. Later, the requirements may be refined or modified by interaction between the design and system-engineering groups and by direction from outside authority. Functional analyses and requirement allocations will also be performed by the system engineering group. These analyses will specify critical time periods and availability of time for repair or reset and indicate the impact of changes in threat scenario.

c. The system engineering group will thus identify each critical function and the system or subsystem that must perform the function and will allocate performance requirements. Identification of specific items of equipment and physical layout may require

Table 2-1. Shock Isolation Objectives

Functional Requirement			Applicability: Prime-Mission Materiel	Applicability: Prime-Mission Personnel
Mitigate	Airblast	To	TR	TR
	Near-surface ground shock	levels	TR	TR
	Deep underground ground shock	tolerable	TR	TR
	Ejecta/debris impact	to	TR	TR

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TR = transattack

tradeoff studies. Until these studies are completed, some critical equipment will be identified only generically by subsystem or function.

d. While the identification of critical functions will be specific to each application, several general catagories can be identified. These general categories are listed below:

—Power Supply
 Power generation
 Energy storage
—Power Distribution
—Heating, Ventilating, and Air Conditioning
—Life Support
 Domestic water supply
 Sewage disposal
 Lighting
 Air quality control
—Communications
 Telephone
 Telegraph
 Radio
—Data Processing
 Data acquisition and transmission
 Data processing and storage
—Prime Mission Materiel
—Personnel

2-3. Description of facility environments.

a. A shock flow path can be traced from the point of burst, in a free-field environment, to the facility structure and to the equipment and personnel inside the facility. The characteristics of the shock change as it moves along this path. There are complex interactions between the facility structure and the media (air and ground) transmitting the shock. There are further interactions within the structure itself, and between the structure and equipment/personnel.

b. The facility response to nuclear weapon effects produces the motions input to the equipment subsystems and personnel. The response varies at different locations within the facility. These varied responses are called the "facility environments." The facility environments are described with the help of the facility designer.

c. Most structures are very complex in their behavior under dynamic loads. The facility environments can be determined or specified in various ways, but most will necessarily involve simplifying assumptions. The choice of the method of analysis is often related to the importance of the equipment item and its position within the structure. For example, a critical item of equipment, located in a position where its allowable motions are restricted, would require a detailed assessment of input motions. In other cases, rougher approximations may be sufficient.

d. The severity of a facility environment is usually expressed by a response spectrum of the motion expected to occur at a structure/equipment interface. The response spectrum is the peak response of single-degree-of-freedom systems to dynamic inputs, usually plotted as accelerations, velocities, or displacements, all as functions of frequency. It is also referred to as shock spectrum and is discussed further in paragraph 3-7.

e. Other methods of measuring environmental severity include a time history showing, for example, acceleration at the structure/equipment interface as a function of time, and a Fourier magnitude spectrum, in which the magnitude of the Fourier transformation of the input force time history is plotted as a function of frequency (see para. 5-2).

2-4. Determination of equipment/personnel tolerances.

a. As the shock environment is expressed in terms of a response spectrum, it is appropriate to express the shock tolerance of equipment in the same manner. A shock tolerance spectrum, or fragility, of an equipment item is the spectrum of motions the item is expected to survive or has survived. Like the input response spectrum, the tolerance spectrum is expressed in terms of acceleration, velocity, or displacement, all as a function of frequency.

b. Another means of expressing shock tolerance is peak allowable acceleration. However, this measure does not provide information on natural frequencies of the equipment, which are important in shock isolation system design. The preferred method of presenting shock tolerance is, accordingly, the shock tolerance (or fragility) spectrum.

c. Although shock fragility information may significantly affect shock isolation design decisions, such information is not commonly available. There are many uncertainties because of limited data and natural variability of equipment. Shock-qualification data available for the equipment used in existing weapons systems are usually developed on a pass/fail basis. Such data do not define true fragilities. For commercially available equipment (facility equipment) few fragility data are available. Fragility for this class of equipment is commonly estimated from dynamic response analyses of simplified models or from a review of commercial/industrial applications of similar equipment. In some instances, dynamic tests have been conducted.

d. Estimates of human shock tolerance are based mainly on tests related to airborne and motor vehicle environments. The effects on humans depend on magnitude, duration, frequency, and direction of the motion, as well as body position at the time of the disturbance. Human shock tolerance is based on threshold levels for discomfort, pain, loss of balance, and injury. Analysis of accident injury data (not the result of testing) has provided additional means of defining upper bound levels for serious injury/death.

2-5. Comparison of environments and tolerances.

a. With descriptions of both the shock environment and tolerance levels in spectrum form, the risk of shock damage can be evaluated by straightforward comparison. If the expected shock for the facility environment exceeds the shock tolerance of personnel, a shock isolation system is required. If the shock input exceeds the tolerance of equipment, the equipment can either be made more rugged to increase its shock tolerance or it can be shock isolated. There are practical limits to how rugged equipment can be made, however, and the cost of making equipment more rugged may exceed that of a shock isolation system.

b. If the shock environment does not exceed the shock tolerance of the equipment, the equipment can be hard-mounted to the structure. This situation is illustrated in figure 2-1. The upper curve shows the shock tolerance or fragility established by testing. The lower curves show a best estimate and conservative estimate for the shock environment. It is seen that even the conservative estimate is well below the equipment shock tolerance.

2-6. Shock isolation system design procedures.

a. The four steps described determine the environment and tolerances of critical equipment and personnel and identify those elements needing shock isolation. Shock isolation system design can then begin as follows:

(1) Derive performance requirements for the shock isolation system from the characteristics of the anticipated shock environments and shock tolerances.

(2) Select the basic shock isolation configuration on the basis of the performance characteristics of generic configurations—pendulum, base mounted, and hybrid.

(3) Assign isolation units: isolate by individual item, by platform-mounting groups of items, or by supporting all the equipment on a major structure within a protective shell.

(4) Select types of isolator elements and estimate their preliminary sizing.

(5) Examine performance capability, based on simple models of ideal isolator elements (massless springs, pure dampers, single-degree-of-freedom elements). Look for cost/performance tradeoffs, potential performance optimization, and other factors such as off-the-shelf availability.

(6) In the final design phase, assess performance

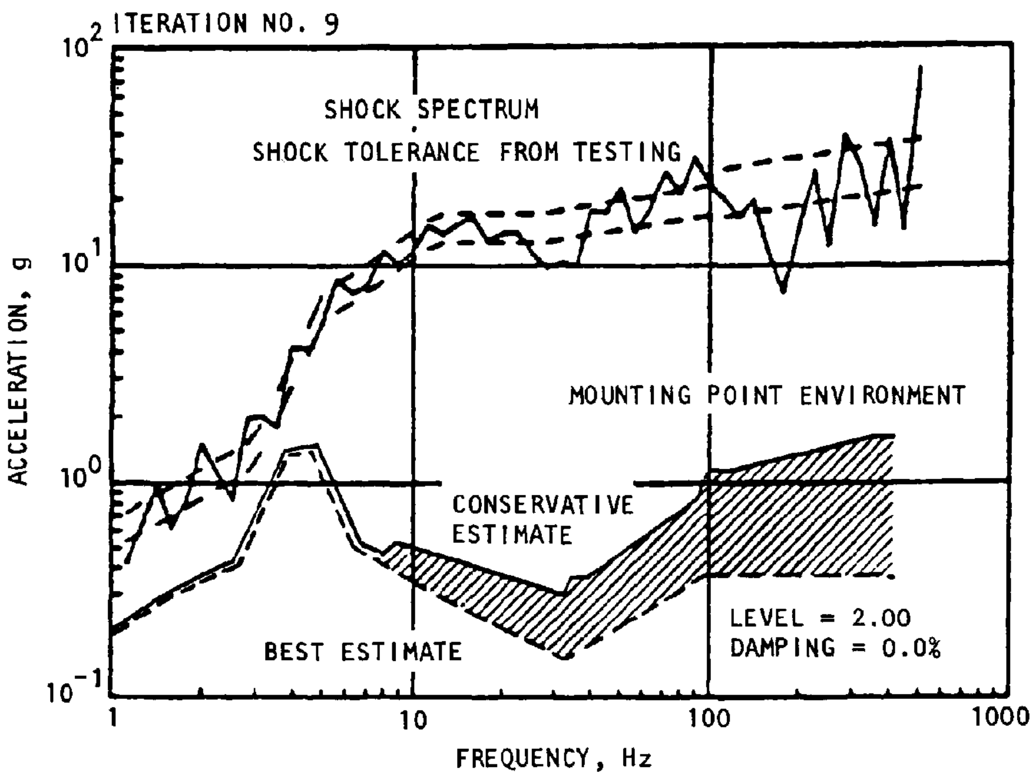

Figure 2-1. Comparison of Maximum Experimentally Observed Vertical Environmental Shock Tolerance and Local Environments (Conservative and Best Estimate Levels)

using more complex models and analysis methods. Although assessment in some respects closely resembles system verification activities described in TM 5-858-6, it is an essential part of the design process.

b. The following sections of this volume discuss the general system design procedures in more detail, beginning with the description of facilities environments. This assumes that critical functions have already been identified; such identification is specific to each application.

CHAPTER 3

DESCRIPTION OF FACILITY ENVIRONMENTS

3-1. Introduction.

a. Facility environments depend on a chain of events that begins with the primary shock environment caused by nuclear detonation. The airblast and ground shock making up the primary shock environment differ in their relative contribution to the facility environment, depending on the location of the structure. In this chapter, the characteristics of airblast and ground shock are described first to provide a basis for understanding and describing facilities environments.

b. The airblast and ground shock interact with the facilities structure, resulting in interior shock environments. Structure analysts using approximate transfer characteristics ascertain the environment transmitted from the exterior structure surface to the interior. The type of information passed to the shock isolation system designer is described briefly.

c. Within the facility, loads are transmitted from structures to equipment and personnel. To determine these loads, a series of ratios is useful to the shock isolation system designer. The ratios include dynamic stiffness, impedance, dynamic mass, compliance, mobility, and inertance. Another type of useful ratio is the transfer function. These ratios are defined and their use in analysis is briefly reviewed.

d. Some actual protective structures are then examined using the ratios defined, particularly the impedance function. Impedance data obtained using a scale model are also presented.

e. Facilities environments are commonly measured or estimated by means of a shock spectrum, which is a plot of the maximum response to a given shock input. The concept of shock spectrum is reviewed.

3-2. Primary shock environments.

a. Shock from a nuclear explosion may reach a structure both from airblast overpressure and from stress waves or shock signals through the ground. The relative importance of airblast or ground shock depends on the location of the structure with respect to the ground surface. Structures are usually classified as aboveground, flush, shallow buried, or deep buried.

b. Aboveground structures will be driven by free-field overpressure strongly altered by the profile of the structure, including the effects of reflected shocks on exposed surfaces. Ground motion plays an additive role in determining the shock environment. Flush structures will be driven by both the free-field overpressure and airblast induced ground-shock signals. The relative importance of these two loads will depend on the size and shape of the structure and the properties of surrounding soil. Shallow and deep-buried structures will be affected only by ground-shock signals—airblast induced for the former and crater- (or direct-) induced for the latter. Table 3-1 indicates the primary shock environment for each of these locations.

3-3. Description of airblast and ground shock.

a. Input from prior analyses. The shock isolation system (SIS) designer is primarily concerned with the shock environments at particular locations within a protective structure. The description of these environments depends on prior analyses of free-field effects, structure-media interaction, and structural loading discussed in TM 5-858-2 and TM 5-858-3. The SIS designer, however, is better able to perform shock-isolation design if he is aware of some of the basic characteristics of airblast and ground-shock waves.

b. Relation between airblast and ground-shock waves. The airblast overpressure and ground-shock waves are illustrated in figure 3-1. The air-induced ground wave consists of compression and shear waves. If these waves trail the overpressure wave, the airblast is superseismic. If one or both of the waves lead the overpressure wave, the airblast is transseismic or subseismic. In this case, the leading wave is called an "out-

Table 3-1. Shock Environment for Different Structure Locations

Classification	Overpressure/Stress Range	Primary Shock Environment
Aboveground	0 to 10's psi	Airblast
Flush	10's to 1000's psi	Airblast and airblast-induced ground shock
Shallow Buried	100's to 1000's psi	Airblast-induced ground shock
Deep Buried	<100,000 psi	Crater or direct-induced ground shock

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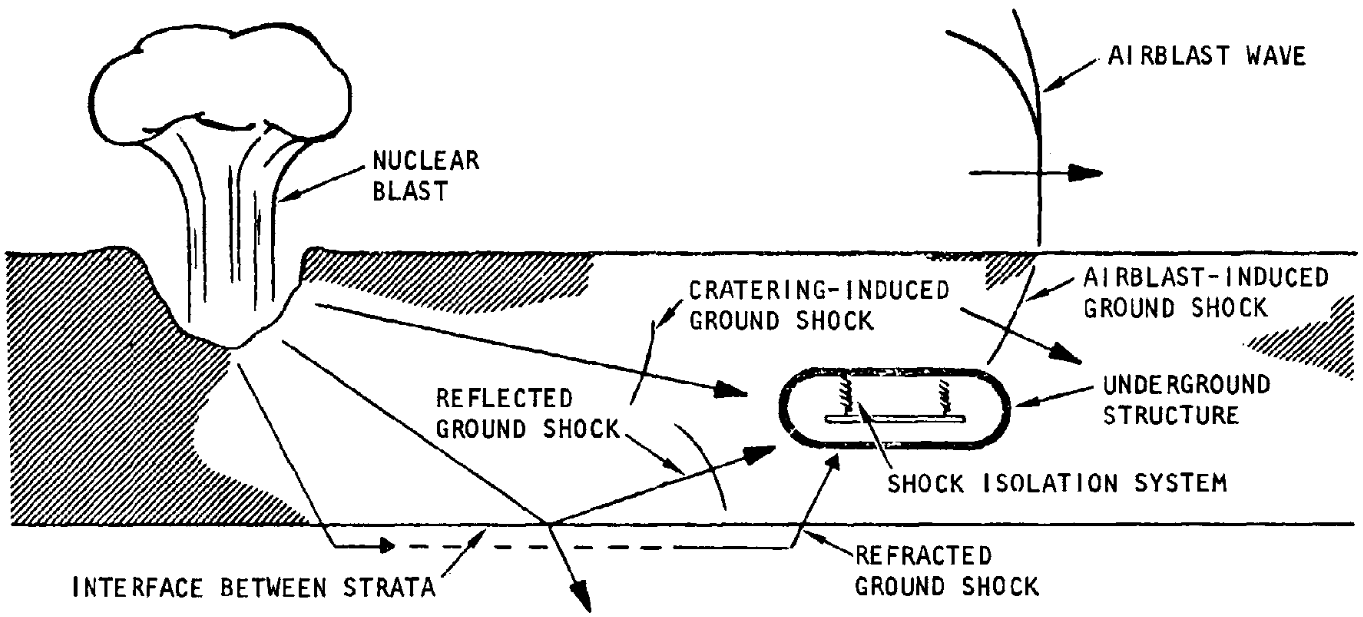

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Figure 3-1. Transmission of Waves Resulting from Nuclear Explosion runner." The crater- (or direct-) induced wave is completely underground.

c. Characteristics of airblast pulses.

(1) Typical overpressure pulses are shown in figure 3-2. These pulses pertain to exposed surfaces of flush structures. Note that the abscissa represents time, with the distance from burst fixed. A method of treating the important uncertainties associated with these measurements is given in TM 5-858-2.

(2) The frequency content of the airblast waves shown in figure 3-2 is important to the SIS designer, as it will strongly influence shock isolation design. The frequency content of the waves, modified by the velocity of the waves as they engulf a structure, constitutes the frequency amplitude forcing function that drives the structure. These frequency-amplitudes are either magnified or attenuated by the frequency selective characteristics of the protective structure as the waves are transmitted internally to mounting locations of sensitive equipment.

(3) The frequency content of the waves can be determined by Fourier analysis. Ideally, free-field overpressure is described by the equation

$$P(t) = P_{so}(ae^{-\alpha t} + be^{-\beta t} + ce^{-\gamma t})\left(1 - \frac{t}{D_P^+}\right) \qquad (3\text{-}1)$$

where

P_{so} = Peak overpressure, psi
$a,b,c,\alpha,\beta,\gamma$ = Constants (see Brode, 1964)
D_P^+ = Positive phase duration of overpressure, sec
t = Time, sec

A Fourier transformation can be performed on this equation. Using values of the constants appropriate to the 7.5 psi and 750 psi curves of figure 3-2 yields the frequency content of these two curves, shown in figure 3-3. Note the higher frequency content of the shorter-duration, higher-pressure blast wave.

(4) In addition to overpressure, dynamic pressure contributes to structural loading. Dynamic pressure arises from the mass flow of air (wind) behind the airblast front. The ideal dynamic pressure is given by:

$$Q_o(t) = Q_s(de^{-\delta t} + fe^{-\phi t})\left(1 - \frac{t}{D_u^+}\right) \qquad (3\text{-}2)$$

where

Q_s = Peak dynamic pressure, psi
d,f,δ,ϕ = Constants (Brode, 1964)
D_u^+ = Positive phase duration of dynamic pressure, sec
t = Time, sec

(5) Free-field overpressures are modified by structural shape, including reflection surfaces and drag forces. The total pressure acting on simple exposed structures is:

$$P_t(t) = R(t) + P(t) + C_D Q_o(t) \qquad (3\text{-}3)$$

where

R(t) = Reflected pressure, psi
C_D = Drag coefficient

The form of $P_t(t)$ is illustrated in figure 3-4. The maximum amplitude of $P_t(t)$ was obtained from reflection factor data presented in Chapter 10 of TM 5-858-3 for various overpressure domains and angles of incidence of the blast on exposed surfaces of structures.

(6) Uncertainties: There are numerous sources of uncertainty about airblast peak pressure and positive phase duration (time during which the pressure is greater than atmospheric). Such sources include, for example, height of burst, miss distance, and weapon yield. Uncertainties from the various sources are combined to provide uncertainty measures for overpressure, dynamic pressure, and total pressure as follows:

(a) For overpressure, if the high-frequency component of the pulse is of primary interest, uncertainty is taken as that associated with peak overpressure; i.e., $\Omega_P = \Omega_{P_{so}}$, as defined in TM 5-858-2. For the low-frequency component, uncertainty is taken as that associated with overpressure impulse; i.e., $\Omega_P = \Omega_I$.

(b) For the high-frequency component of the dynamic pressure,

$\Omega_{Q_o} = \Omega_{Q_s}$, and for the low-frequency component, $\Omega_{Q_o} = \Omega_I$.

(c) For total pressure, the square of the uncertainty is given by

$$\Omega_{P_t}^2 = \frac{1}{P_t^2} P^2\Omega_P^2 + C_D^2 Q_o^2 \Omega_{Q_o}^2 \qquad (3\text{-}4)$$

Equation 3-4 is a time-dependent relationship. For high-frequency systems, $\Omega_{P_t}^2$ is evaluated at a time frame near the occurrence of peak overpressure or dynamic pressure. For low-frequency systems, $\Omega_{P_t}^2$ is evaluated over the entire duration of the pulse.

d. Characteristics of ground-shock signals.

(1) Ground-shock signals play a role in driving structures at all depths. TM 5-858-2 defines ground shock as airblast-induced if the structure is located in a region less than 30 deg below the horizon and crater-induced if it is located below that angle.

(2) Speed of the airblast pulse varies directly with the square root of the pressure. Therefore, near ground zero the speed of the airblast pulse will exceed that of the ground shock pulse, and the structure will be in a superseismic region. As the distance of the structure from ground zero increases, the structure will be in a transeismic and then in a subseismic region. In these cases an outrunning ground shock will hit the structure before the airblast pulse. TM 5-858-2 takes these factors into account in presenting ground-shock characteristics as a function of range to ground zero, site geology, and depth of structure.

(3) The amplitudes of free-field acceleration, velocity, and displacement for soil and rock sites are

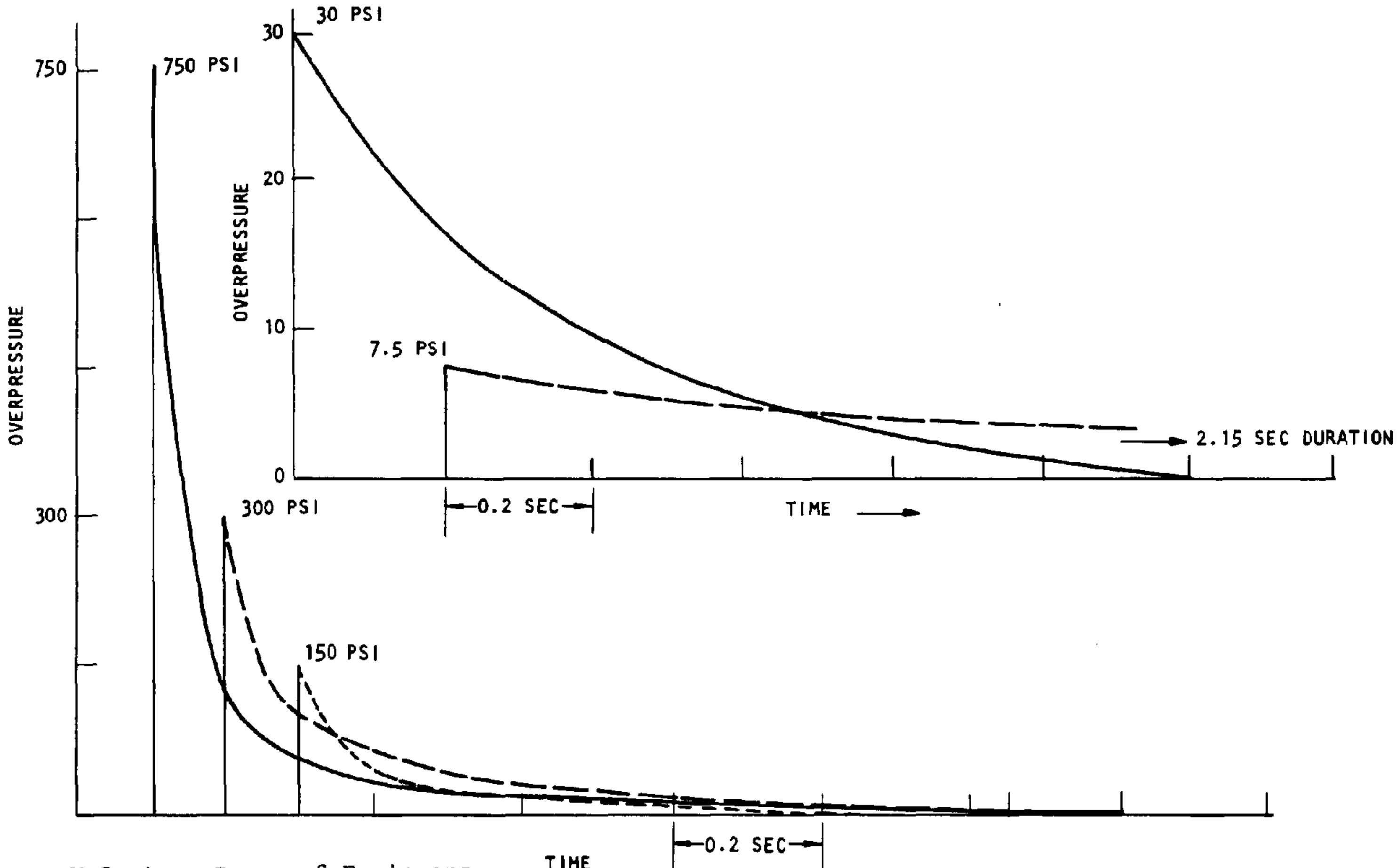

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Figure 3-2. Typical Airblast Overpressure Pulses (Theoretical One-Megaton Contact Burst)

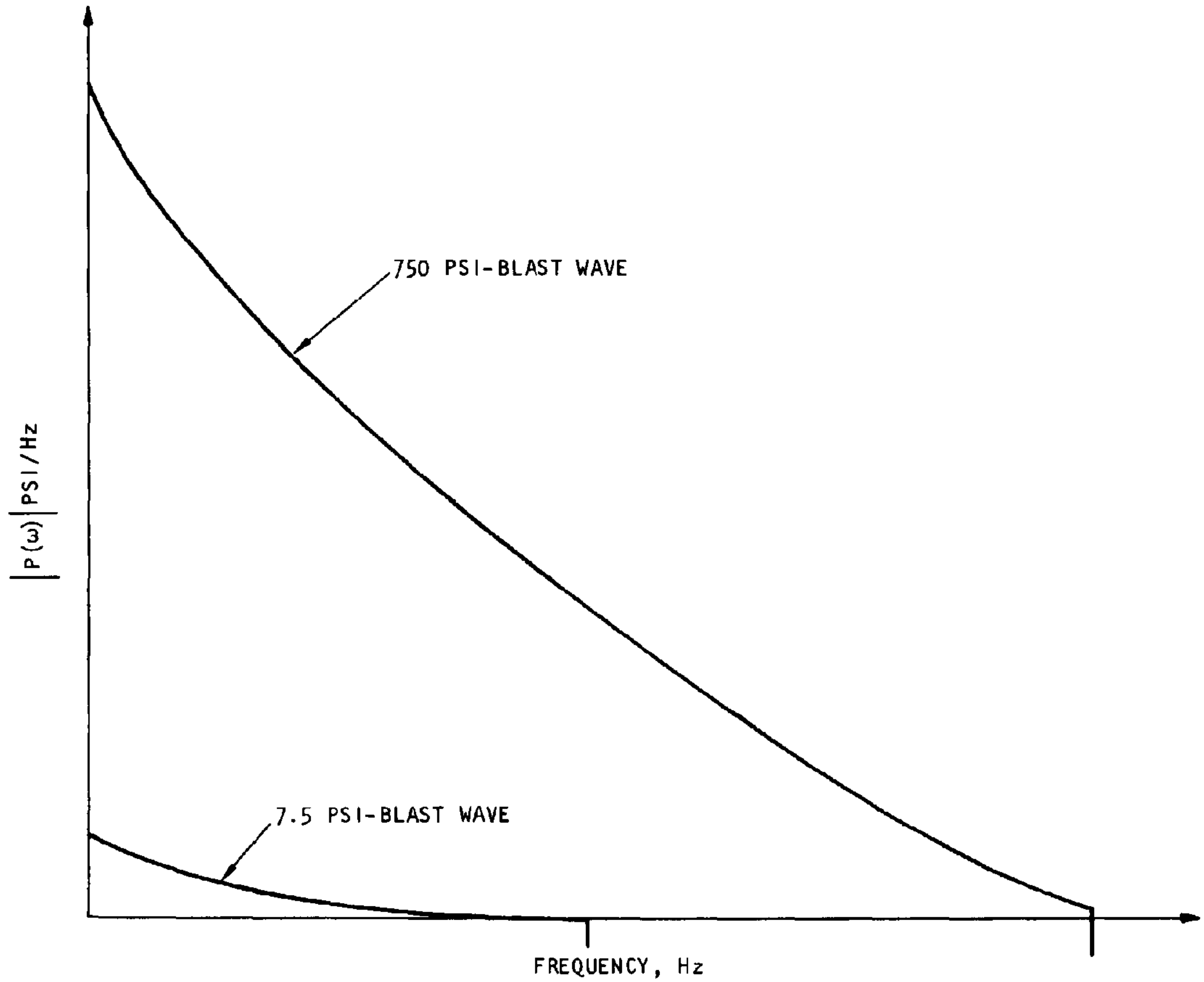

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Figure 3-3. Fourier Magnitude Showing Frequency-Amplitude Contents of 7.5 psi and 750 psi Blast Waves from Figure 3-2.

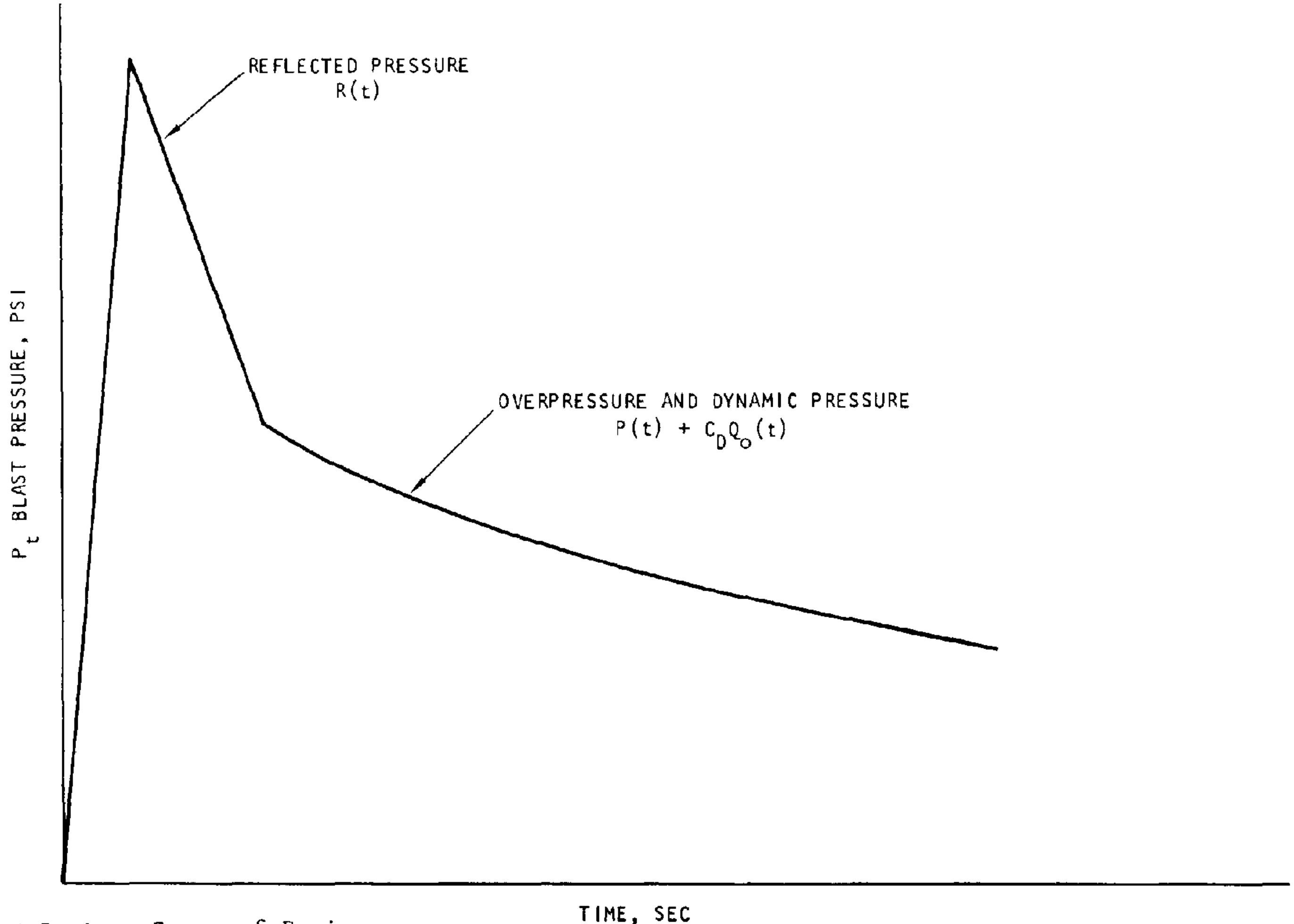

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Figure 3-4. Typical Ideal Total Pressure Pulse on Building Front Face (Related Pulses for other Surfaces) presented in chapters 8 and 9 of TM 5-858-2, together with their associated uncertainty bounds. Also, pulse shapes for soil and rock sites are presented as a function of range and depth in TM 5-858-2. Those data will not be reproduced here.

3-4. Definition of interior shock environments.

a. Free-field environments composed of airblast (on exposed structures) and ground shock interact with the structure, generating a loading profile specific to the structure and the characteristics of the ground media. For exposed structures, complicated airblast loading and drag result from transient aerodynamic engulfment. These interactive loads are transmitted through the protective structure and undergo frequency-selective amplification and attenuation because of the dynamic properties of the structure.

b. In the very early phases of system design, approximate transfer function characteristics can be used to ascertain the environment transmitted from the exterior structure surface to the mounting points of isolated or nonisolated equipment. The results of these analyses using approximate transfer characteristics are given to the shock isolation designer by the structural designer as an important input to his design effort.

c. Two methods are used to determine the approximate transfer characteristics. The standard procedure is to use numerical methods involving the computer programs and finite element codes described in TM 5-858-3.

d. The second method for determining transfer characteristics is to review other protective structures and their associated shock environments for application to the particular system. Scaling and extrapolation of environmental predictions, scale model tests, and field test data from other systems can be useful for initial approximations. Existing protective structures have limited geometric configurations in the form of cylinders, arches, and box shapes. These shapes often permit reasonable scaling due to their similarities to the shapes of new systems under analysis.

e. Table 3-2 lists systems, tests, and organizations that may be reviewed for possible applicable shock environment information. A considerable amount of data from some of these sources is presented in appendix A.

Table 3-2. Sources of Applicable Shock Environment Data

Protective Systems

Minuteman, Minuteman Upgrade, MX System, Safeguard System, Titan System, NORAD System, Sanguine

Controlling Organization

U.S. Air Force Space and Missile Systems Organization; U.S. Army Ballistic Missile Division; North American Defense Command, Defense Communications Agency, Defense Nuclear Agency, U.S. Navy.

Table 3-2. Sources of Applicable Shock Environment Data—Continued

Associated U.S. Government Laboratories/Agencies

U.S. Air Force Weapons Laboratory; U.S. Army Waterways Experiment Station; U.S. Navy Civil Engineering Laboratory; U.S. Army Corps of Engineers, Omaha District; U.S. Army Corps of Engineers, Huntsville

Field Tests

1. HEST tests, Minuteman System
2. High Explosive Events; Distant Plain, Prairie Flat, DIAL PACK, Dice Throw, Mixed Company, Middlegust (see test director's report for applicable experiments)
3. Impedance tests, Safeguard System

U.S. Army Corps of Engineers

3-5. Transmission of structural loads to equipment/personnel.

a. The ratios of various quantities are fundamental to the analysis and design of shock isolation systems. For a simple static system, a well-known ratio is the spring stiffness constant, k, given by

$$k = \frac{\text{Static force}}{\text{Static displacement}}$$

For dynamic systems that are the subject of shock isolation analyses, this ratio becomes dynamic stiffness. Dynamic stiffness and other analogous ratios are basic tools for the shock isolation designer.

b. Dynamic systems by definition involve velocity and acceleration, in addition to displacement. Using dynamic force together with displacement, velocity, and acceleration, six related ratios may be defined that are helpful in characterizing dynamic systems:

$$\frac{\text{Force}}{\text{Displacement}} = \text{Dynamic Stiffness}$$

$$\frac{\text{Displacement}}{\text{Force}} = \text{Compliance}$$

$$\frac{\text{Force}}{\text{Velocity}} = \text{Impedance}$$

$$\frac{\text{Velocity}}{\text{Force}} = \text{Mobility}$$

$$\frac{\text{Force}}{\text{Acceleration}} = \text{Dynamic Mass}$$

$$\frac{\text{Acceleration}}{\text{Force}} = \text{Inertance}$$

In these ratios, the force is the input force on a structure and displacement, velocity, and acceleration are measured at the same point or other points of interest in the structure.

c. Another important series of ratios is defined by the ratio of output force to input force or of output motion to input motion, where "motion" may be displacement, velocity, or acceleration. "Input" refers to one part of a structure, where the force is applied, and "output" to another part of the structure. Such a ratio is called a "transfer function." The utility of this function is that if the base of a linear structure is put into motion by a blast or earthquake, multiplying this base motion by the transfer function produces the motion at another part of the structure.

d. Symbolically, the transfer function is defined by

$$\frac{F_{out}}{F_{in}} = \frac{X_{out}}{X_{in}} = \frac{\dot{X}_{out}}{\dot{X}_{in}} = \frac{\ddot{X}_{out}}{\ddot{X}_{in}}$$

where F is force, X is displacement, and each dot represents differentiation with respect to time. The reason these ratios are equal is that these ratios are in the frequency domain

$$X = (2\pi f)X = \omega X$$

$$\ddot{X} = (2\pi f)^2 X = \omega^2 X = \omega \dot{X}$$

where f is frequency in Hertz and ω is circular frequency in radians per second. When the input and output quantities are divided to form the transfer function, the ω's cancel.

e. The impedance-type ratios are ratios of unlike quantities and have dimensions, whereas the transfer function is dimensionless. Depending on the type of input, the designer will choose either an impedance-type ratio or a transfer function to perform his analysis. Generally, the designer will be concerned with output motion (not force) at some point in a structure. If the input is a motion, he can use a transfer function. If the input is a force, he can use an impedance-type function. Force ratios are equivalent to motion taken over the same structural path but in the opposite direction.

f. The various ratios are difficult to evaluate if force and motion are expressed as functions of time. They are relatively easy to evaluate, however, if they are expressed as functions of frequency. For this purpose a Fourier transformation can be performed, converting a time history of, say, a force into two functions of frequency, one a magnitude and the other a phase.

g. The use of an impedance-type function can be illustrated as follows: Suppose an input force time history, F(t), and an output acceleration time history, $\ddot{X}(t)$, at some internal point of the structure are available to the analyst, perhaps from a test. A Fourier transformation is performed on each of these time functions to obtain magnitude and phase functions for each, i.e., $|F(j\omega)|$ and $\phi(j\omega)$ for force, and $|\ddot{X}(j\omega)|$ and $\theta(j\omega)$ for acceleration. The inertance can now be calculated as having magnitude and phase of

$$\frac{\ddot{X}(j\omega)}{F(j\omega)} \quad \text{and } \phi(j\omega) - \theta(j\omega)$$

Now an input force of interest, F(t), is postulated (perhaps estimated from a nuclear blast). To determine the output acceleration produced by this force, a Fourier transformation is performed to obtain $|F(j\omega)|$ and $\psi(j\omega)$. Multiplication of $|F(j\omega)|$ by the inertance gives $|\ddot{X}_{out}(j\omega)|$ and addition of the phases gives the phase, $\psi + \phi - \theta$. An inverse Fourier transformation now produces $\ddot{X}_{out}(t)$, the desired output time history of acceleration at the location of interest.

h. The same procedure is used for a transfer function. An input acceleration time history, say, is Fourier-transformed to obtain magnitude and phase as a function of frequency. These are combined with the transfer function in a manner similar to that performed with inertance in the preceding paragraph. The result is the output acceleration magnitude magnitude and phase. An inverse Fourier transformation now produces the desired output acceleration time history.

3-6. Measured impedances and use of models.

a. Examples of measured impedances for a number of protective structures are presented in figures 3-5 through 3-17. These illustrate the magnitude of motions that may be transmitted in a structure and also show the frequency-selective nature of the transfer functions. Protective structures include the following and are ranked in order of highest to lowest impedance:

—Minuteman Silo (flush buried)
—Perimeter Acquisition Radar Building, Safeguard System 1/12-scale model structure (above ground)
—Aircraft Protective Hangarette (soil-covered arch)
—Electrical Distribution Center, Safeguard System (flush buried)
—Sprint Silo, Safeguard System (flush buried)
—Box structure, thick wall, WES Experimental (buried)
—Box structure, thin wall, WES Experimental (buried)

An example of each of the first five protective structure types listed above is illustrated in figures 3-5, 3-7, 3-10, 3-12, and 3-14, respectively. Generally, the figures indicate the location of applied force, output motion, and impedance path. Figures 3-6, 3-8, 3-9, 3-11, 3-13, and 3-15 through 3-21 reflect the measured impedance for the structure types as a function of frequency.

b. Impedance functions in the examples can be generally characterized by alternating resonances and antiresonances over the frequency spectrum. For example, in figure 3-6 and similar figures where impedance is plotted, the bottom points are resonances, and the upper points (spikes) are antiresonances. In figures 3-11 and 3-13, where inertance is plotted, the reverse is true and the spikes are resonances. Bounds for resonance and antiresonance for the example systems are given in table 3-3. This table also presents other sum-

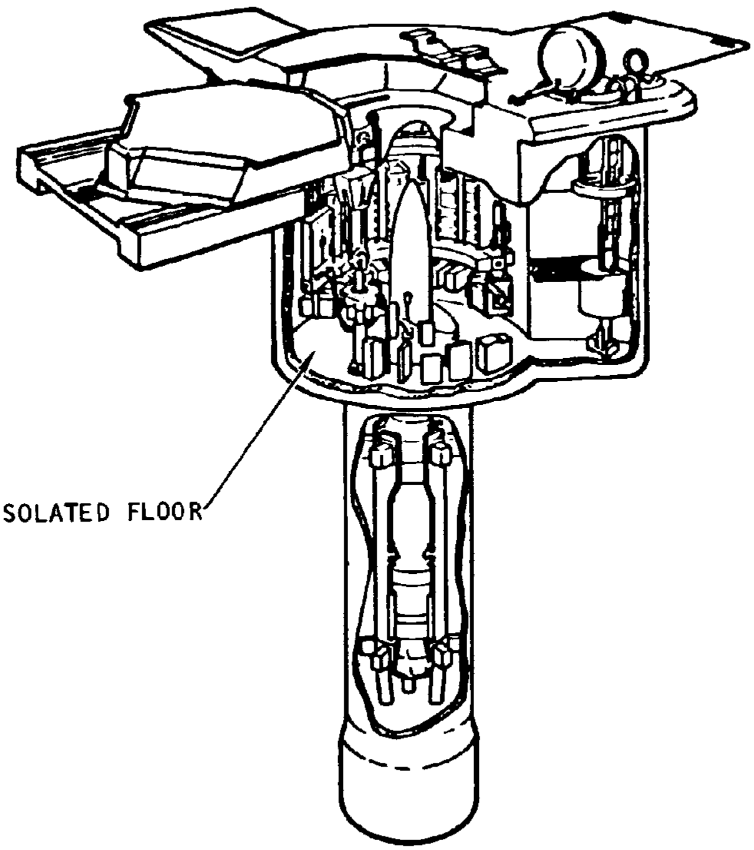

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Figure 3-5. Minuteman System, Transfer Impedance Measurement from Center of Silo Cover to Isolator Attachment Location

mary information from the examples. The numbers in the table indicate the uniqueness of the structures. Most cases of practical interest will be in the range of values found in the table.

where f if frequency and X is displacement. Acceleration is given by

$$\ddot{X} = 2\pi fV = 200\pi \times 10^{-3} = 0.63 \text{ in./sec}^2$$

Similar motion determinations can be carried out for all other frequencies (and the associated magnitudes) that may be contained in the external loading airblast or ground shock (fig. 3-3).

e. Figure 3-15(b) illustrates an impulse function. Impulse is defined as acceleration (or velocity) divided by force as a function of time. Mathematically, it is obtained by performing an inverse Fourier transformation on the impedance function. In general, the more spread out (into higher frequencies) the impedance function, the shorter the impulse function, and vice versa.

f. If a protective facility has a configuration somewhat similar to any of the examples, the system function can be scaled for an initial estimate. Alternatively, a scale model can be constructed. The measured impedances and transfer functions taken from the model can be scaled to prototype dimensions.

g. Figures 3-18 and 3-19 present data obtained both by scaling up from a $\frac{1}{12}$-scale model test and by scaling up impedance measurements on the model. Figure 3-18(a) shows the scaled DIAL PACK model response, measured by direct acceleration records during a 500-ton TNT explosive test. Figure 3-18(b) shows the scaled response, as predicted from impedance measurements on the model combined with the airblast characteristics. Figure 3-18(c) is the same type of calculation as in (b),

c. Figures 3-7, 3-8, and 3-9 illustrate impedance measurements for both a model and a prototype Safeguard Perimeter Acquisition Radar Building. The (a) and (b) plots for each figure illustrate the concept of reciprocity: the drive point and measurement point are interchanged and the impedance plots compared. If the system is linear, these plots will be the same. Note that lines of constant stiffness (force/displacement) and constant mass (force/acceleration) are drawn on the charts. These lines are sometimes helpful in interpreting and comparing charts.

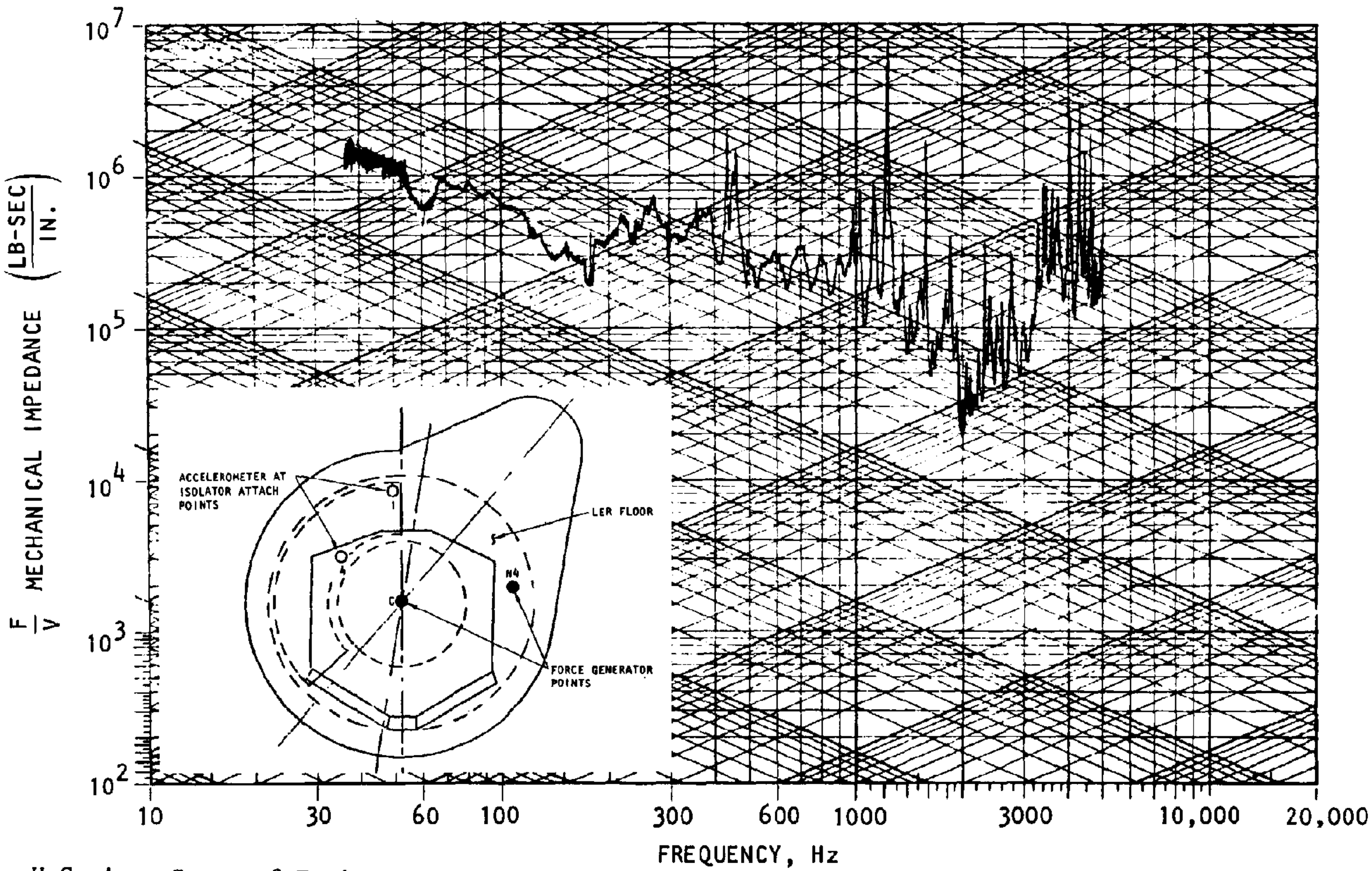

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Figure 3-6. Impedance Plot for 30 Hz to 5000 Hz, Surface of Launch Equipment Room (LER) to Ceiling Isolator Attachment No. 4

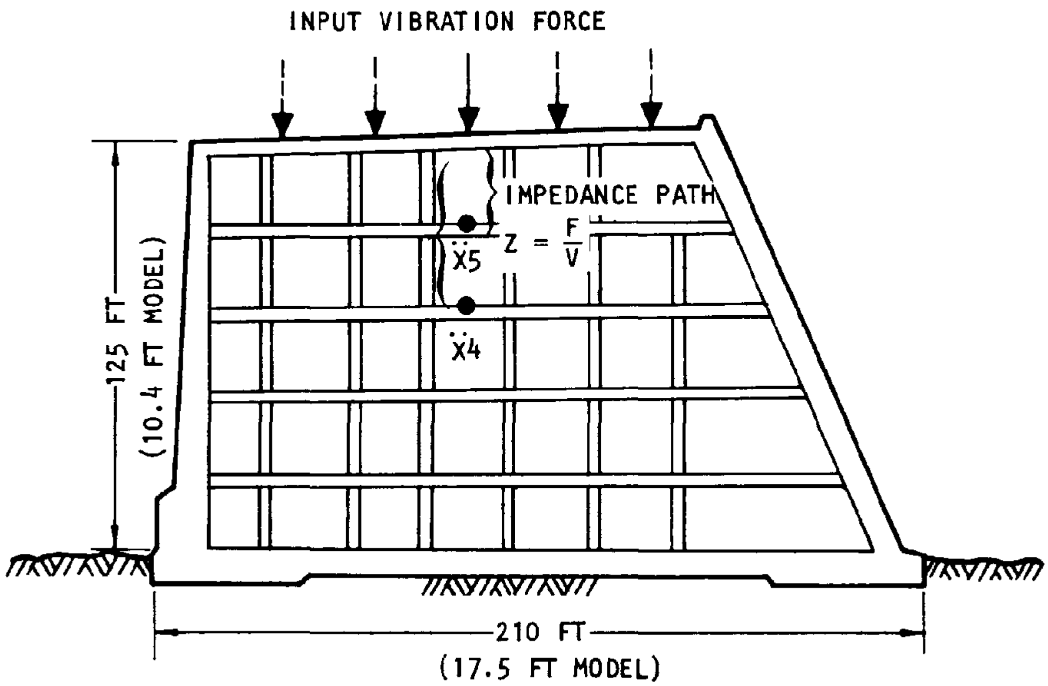


Figure 3-7. Perimeter Acquisition Radar Building (PARB) Safeguard System (Safford et. al, 1977)

d. A single-frequency interpretation of the data in figure 3-6 can be illustrated as follows:

At 100 Hz, impedance is approximately

$$\frac{F}{V} = 5 \times 10^5 \text{ lb-sec/in}$$

where F is force and V is velocity. For a 500-lb force,

$$V = \frac{500}{5 \times 10^5} = 10^{-3} \text{ in./sec}$$

For displacement,

$$X = \frac{V}{2\pi f} = \frac{10^{-3}}{2\pi(100)} = 1.6 \times 10^{-6} \text{ in.}$$

except the impedance measurements were made on a full-size prototype, and the airblast was scaled up accordingly. Figure 3-19 is a plot of the three corresponding Fourier magnitudes of acceleration.

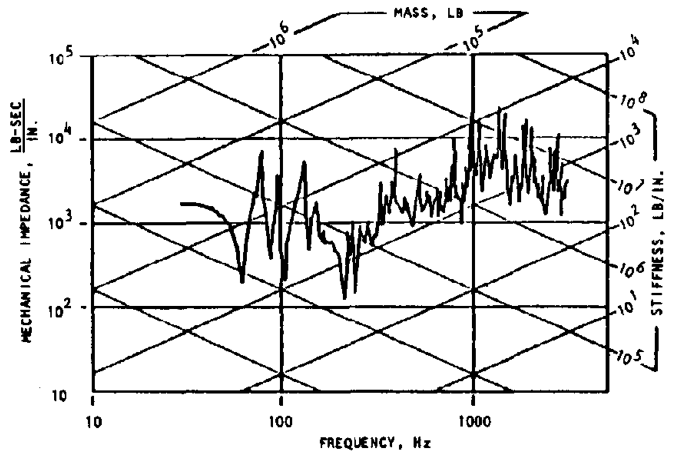
3-7. Description of shock spectrum.

a. Figure 3-20(a) shows an input shock excitation driving a base to which a series of single-degree-of-freedom systems (for example, reed gages) are attached. Each single-degree-of-freedom system is "tuned" to a different frequency. The response can be measured as displacement, velocity, or acceleration relative to the base. Figure 3-20(b) shows how the system is represented as a mathematical model for calculation of the shock spectrum.

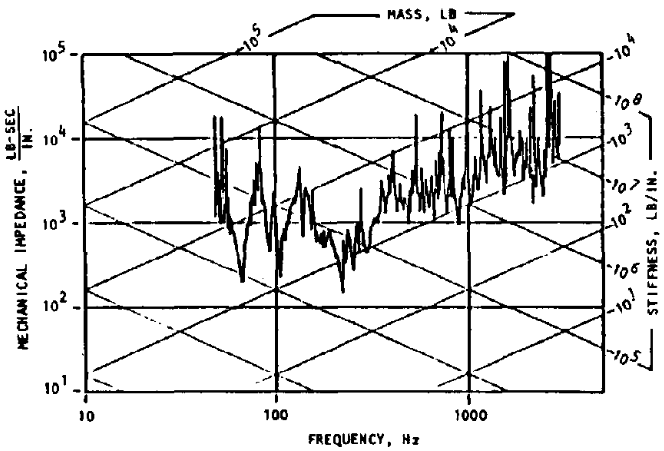
b. If a time history is recorded for the response of each single-degree-of-freedom system, a maximum will be obtained for each system. A shock spectrum is a plot of the maximum responses of the set of single-degree-of-freedom systems to the given shock input, as shown in figure 3-20(c).

c. A common means of presenting a shock spectrum is by means of a four-coordinate logarithmic graph such as shown in figure 3-21. Lines of constant maximum relative displacement and acceleration plot as straight lines with slope +1 and −1, respectively. The plot in the figure shows that, for example, if the maximum acceleration is to be 3 g, the frequency of the shock isolation system should be 3.2 Hz and a maximum relative displacement of 2.8 in. will occur (Ruzicka, 1967).

d. As a measurement tool, the shock spectrum has certain limitations that should be recognized. Different time histories of shock excitation can give approximately the same shock spectra. Therefore, to design effectively, the shock isolation system designer needs some information on time history in addition to the shock spectrum. When used in conjunction with such time history information, shock spectra can be useful tools.

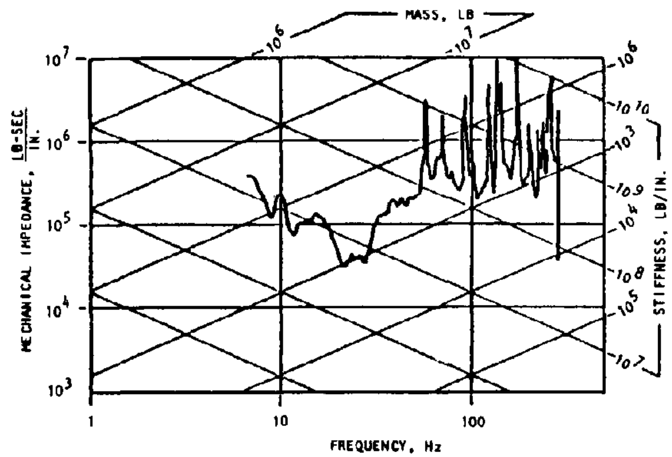


(a) Drive point, roof center;
Measurement point, 5th floor center

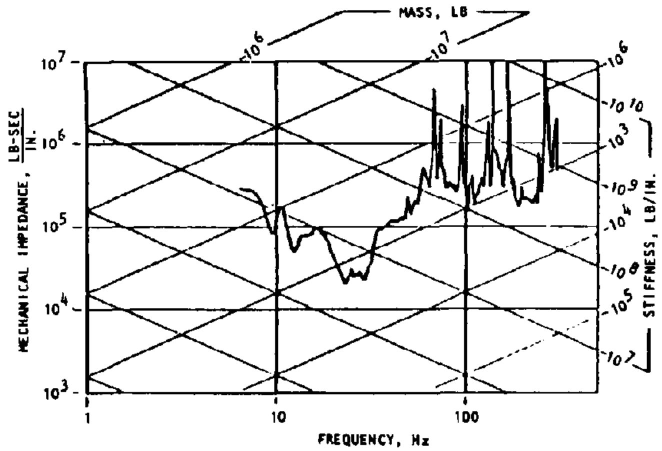


(b) Drive point, 5th floor center;
Measurement point, roof center

FIGURE 3-8. MODEL PARB DATA FOR DETERMINING RECIPROCITY (Safford, et al., 1977b)



(a) Drive point, roof center;
Measurement point, 5th floor center



(b) Drive point, 5th floor center;
Measurement point, roof center

FIGURE 3-9. PROTOTYPE PARB DATA FOR DETERMINING RECIPROCITY (Safford, et al., 1977b)

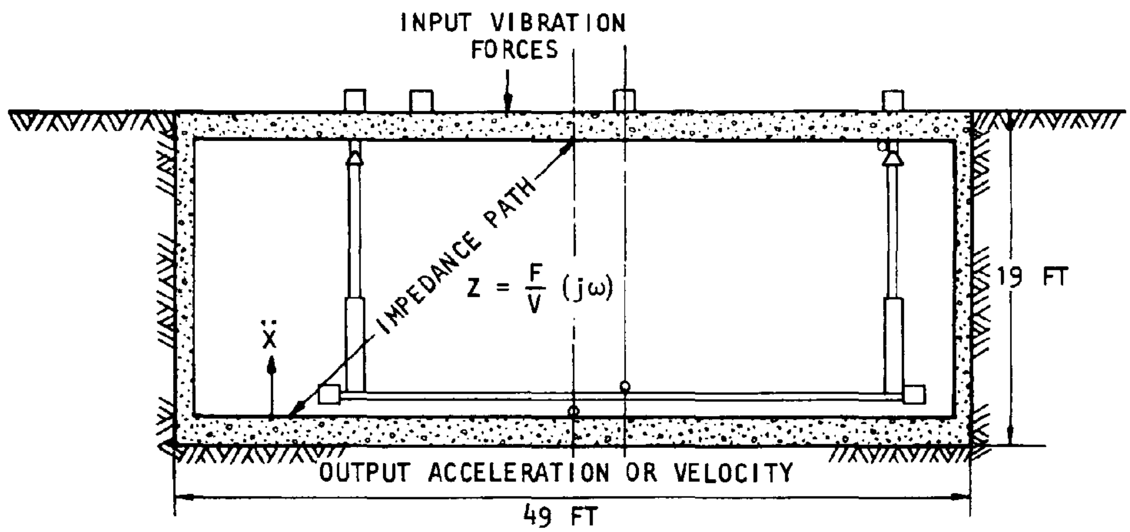


Figure 3-10. Electrical Distribution Center (EDC) Safeguard System (AA, 1978)

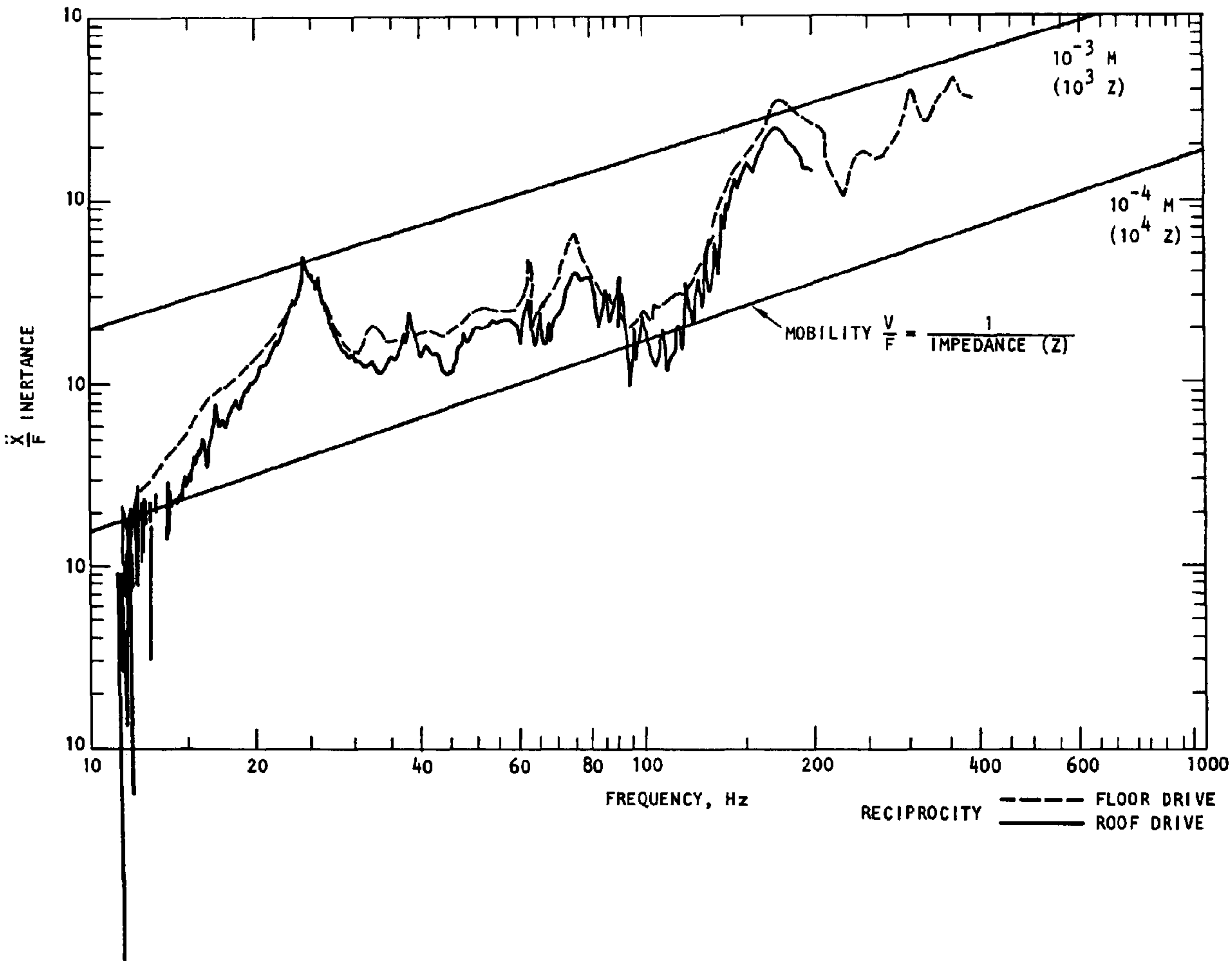


Figure 3-11. Flush-Buried Safeguard Protective Structure, Reciprocal Transfer Inertance Functions from Roof to Floor and Reverse (AA, 1978)

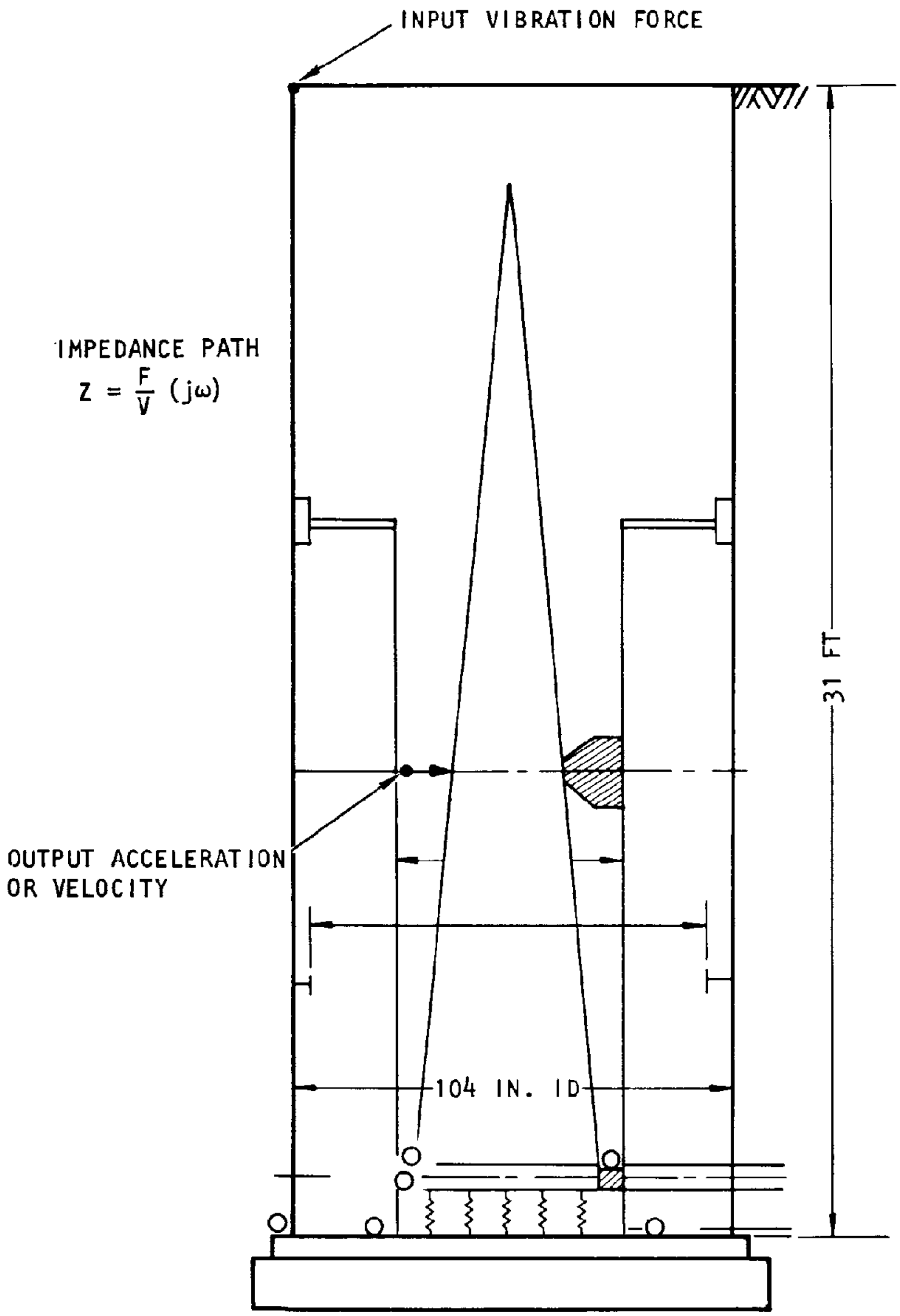


Figure 3-12. Sprint Launch Cell, Safeguard System (AA, 1978)

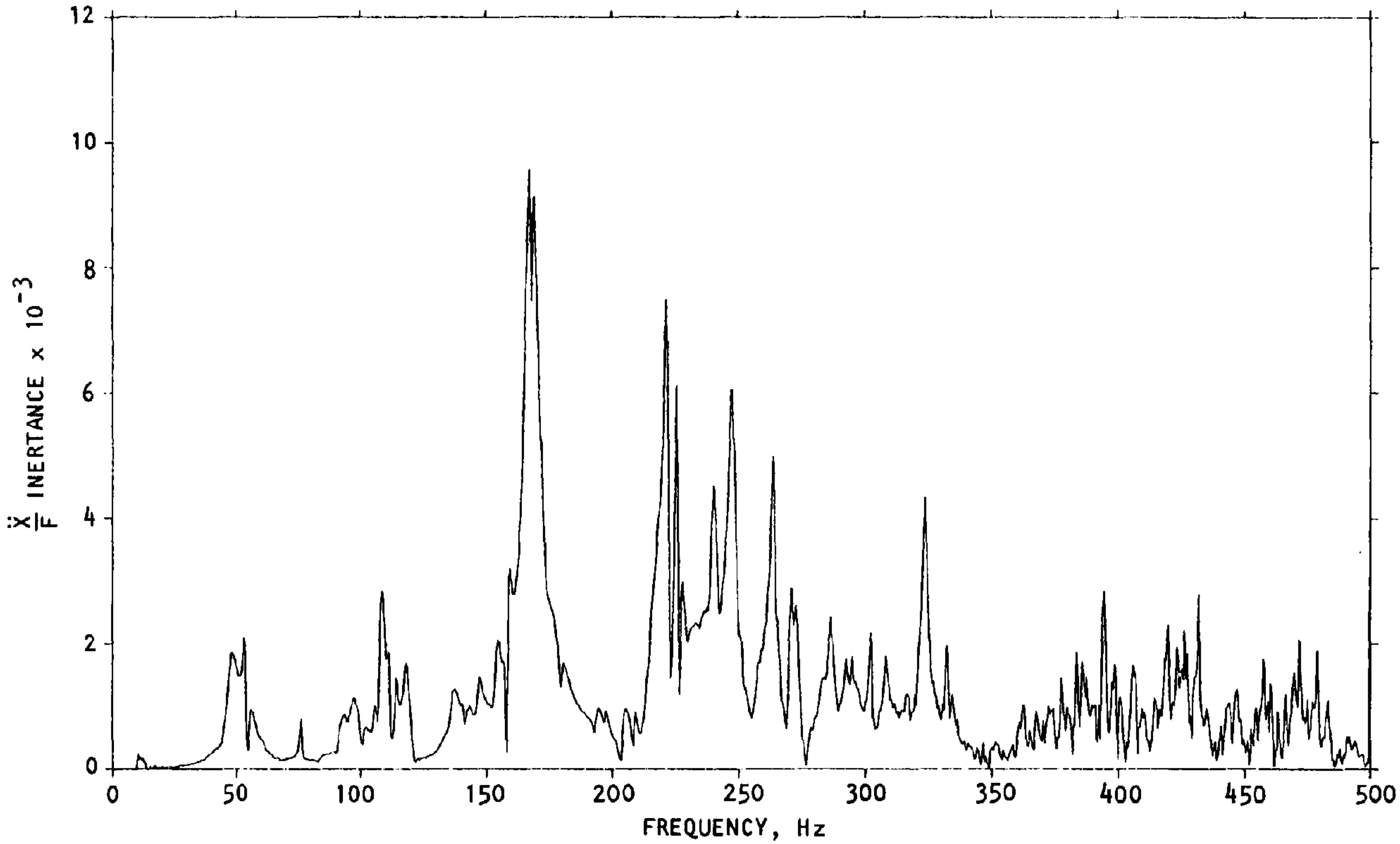


Figure 3-13. Sprint Launch Cell Transfer Inertance, Launch Cell Wall to Upper Missile Support (AA, 1978)

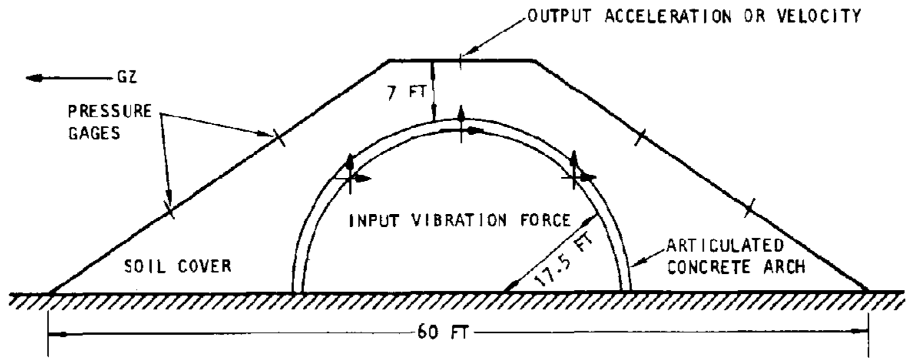
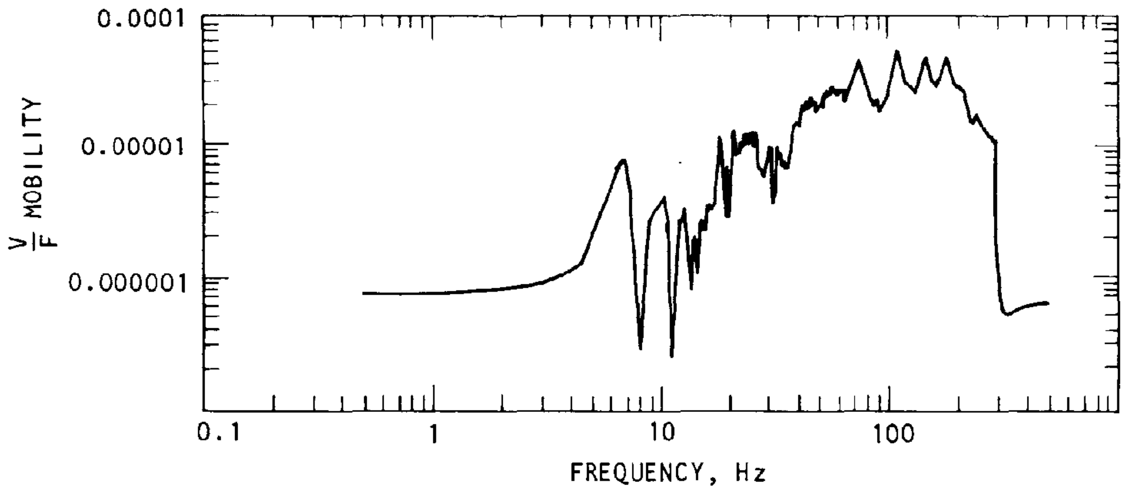
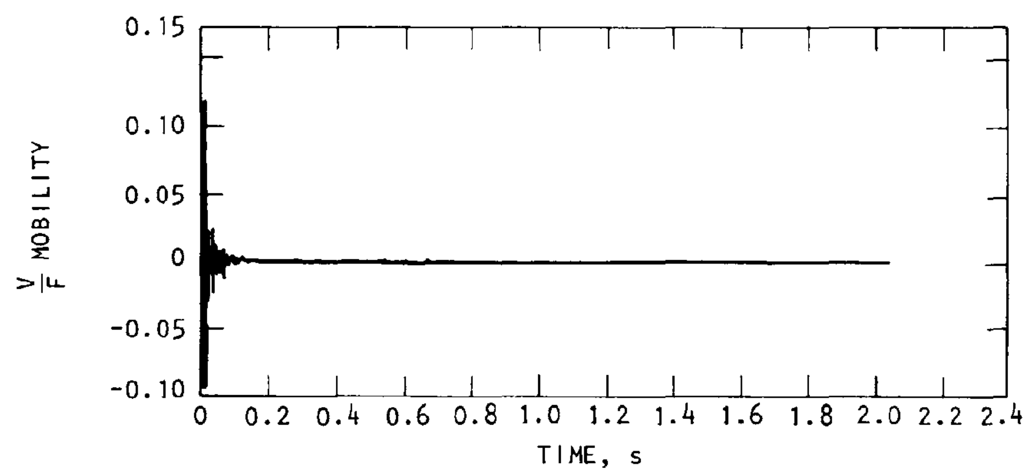


Figure 3-14. Hangarette/Aircraft Shelter Used in Mixed Company Event (500-ton TNT test) and for Impedance Measurements (AA, 1977)



(a) Mobility magnitude function



(b) Mobility impulse function

Figure 3-15. Mobility of Hangarette from Mid-Arch to Soil Surface (Courtesy Waterways Experiment Station) (AA, 1977)

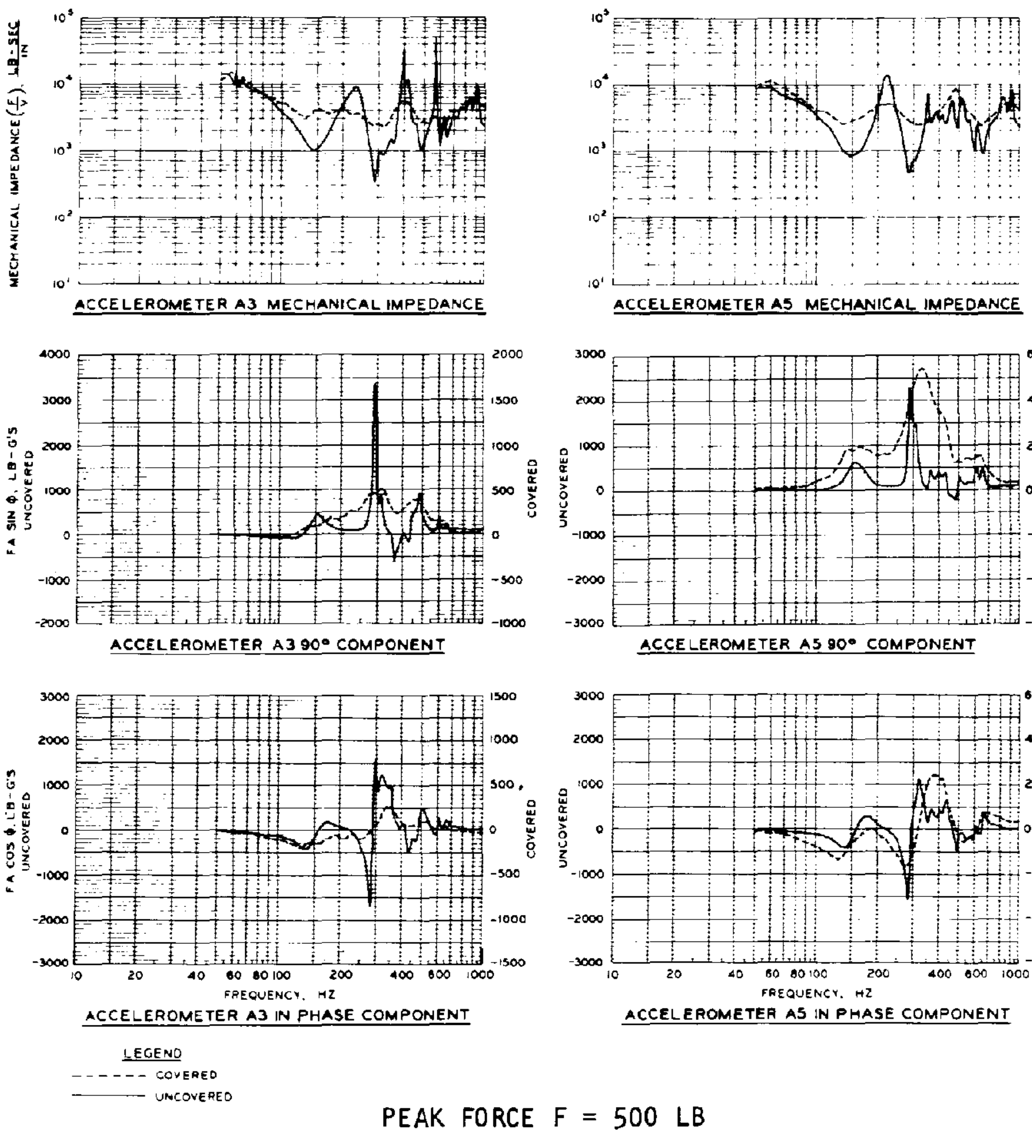


Figure 3-16. Impedance and Quadrature for Rectangular Structure 3B before and after Placement of Backfill Determined from Tests with Vibrators in Phase (Crowson-Kiger, 1977)

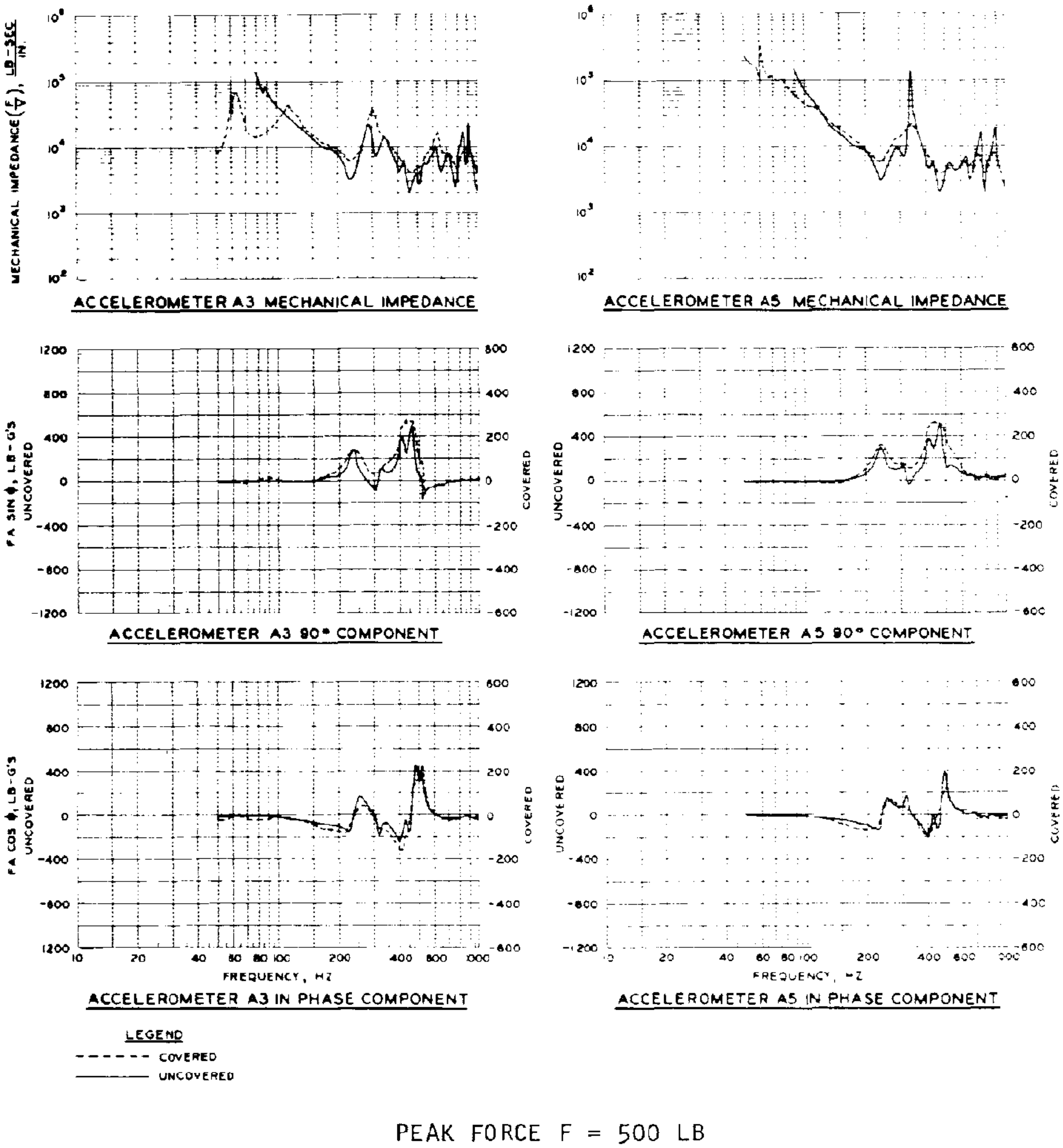


Figure 3-17. Impedance and Quadrature for Rectangular Structure 3D before and after Placement of Backfill Determined from Tests with Vibrators in Phase (Crowson-Kiger, 1977)

TABLE 3-3. ENVELOPE OF RESONANCE AND ANTIRESONANCE IMPEDANCE MEASUREMENT OF PROTECTIVE STRUCTURES

Structure	Impedance Bounds (lb-sec/in.) Resonance	Anti-Resonance	Size (ft)	Construction Material	Siting	Measurement
Minuteman Silo Launch Equipment Room	10^5 40 Hz to	10^6 5000 Hz	X 30'-dia. x 30' deep	Reinforced concrete	Flush to ground	Cover center to isolator attach.
Perimeter Acquisition Radar Building (PARB) SAFEGUARD	4×10^4 5 Hz to	4×10^5 500 Hz	125' x 194' x 210'	Reinforced concrete	Above ground	Roof center to top floor (5th)
Aircraft Protective Hangarette	2×10^4 4 Hz to	3×10^6 300 Hz	17.5' arch radius T' earth cover	Reinforced concrete arch	Earth covered arch	Center of arch to earth surface
Electrical Distribution Center--SAFEGUARD	10^3 12 Hz to	10^4 500 Hz	19' x 49' x 26'	Reinforced concrete	Flush to ground	Roof center to floor center
Sprint Silo, SAFEGUARD	10^3 15 Hz to	10^5 500 Hz	9'-dia., 31' deep, wall 0.04'	Steel	Flush to ground	Silo wall to missile support
Box Structure, Thick Wall Experimental	3×10^3 40 Hz to	5×10^5 1000 Hz	16' x 4' x 4' wall 1.1'	Reinforced concrete	Aboveground	Inside wall to outside wall
	6×10^3 40 Hz to	1.5×10^5 1000 Hz			Fully buried	
Box Structure, Thin Wall Experimental	3×10^2 40 Hz to	2×10^4 1000 Hz	16' x 4' x 4' wall 0.5'	Reinforced concrete	Aboveground	Inside wall to outside wall
	1.5×10^3 40 Hz to	10^4 1000 Hz			Fully buried	Inside wall to outside wall
Perimeter Acquisition Radar Building (1/12 Scale Model)	2×10^2 40 Hz to	2×10^4 3000 Hz	10.4' x 16' x 17.5' wall 0.25'	Reinforced concrete	Aboveground	Roof center to top floor (5th)

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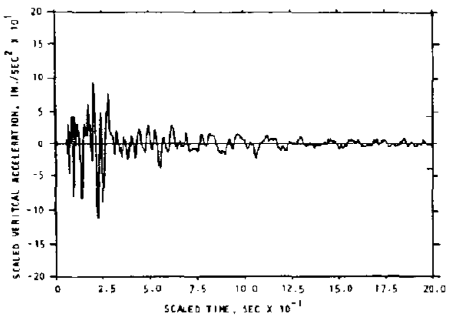
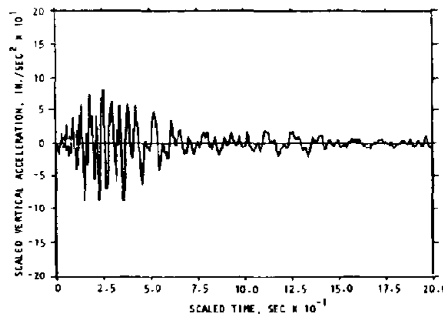
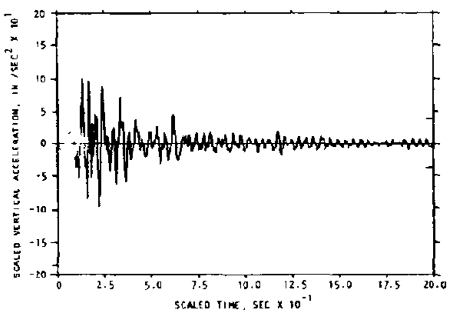


(a) Scaled DIAL PACK model response

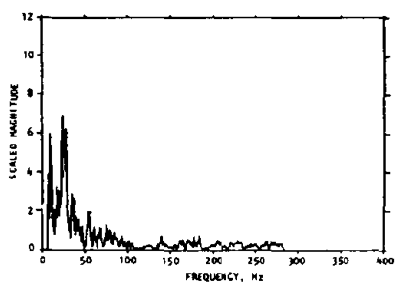


(b) Scaled predicted model response

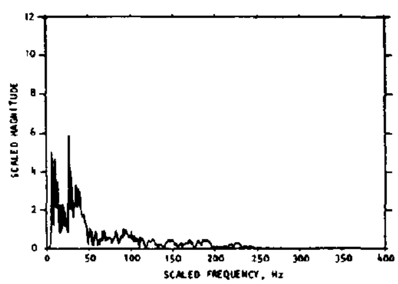


(c) Predicted prototype response

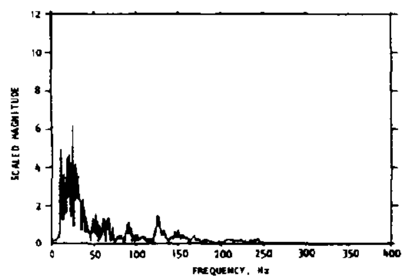
FIGURE 3-18. COMPARISON OF PROTOTYPE AND SCALED MODEL PARB ACCELERATION-TIME HISTORIES: 5TH FLOOR CENTER (Safford et al., 1977b)



(a) Scaled DIAL PACK model response

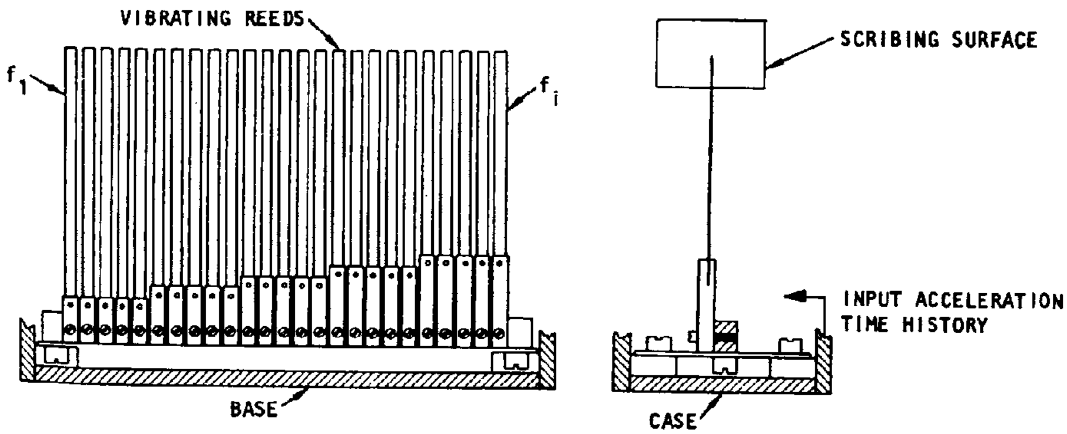


(b) Scaled predicted model response

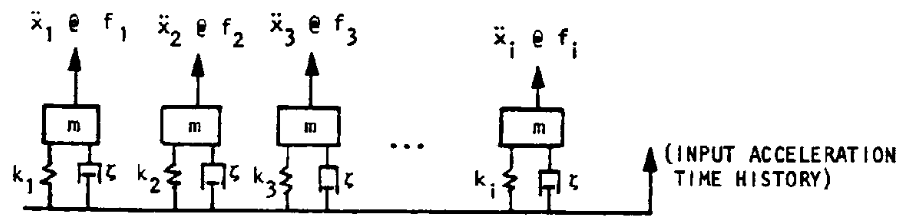


(c) Predicted prototype response

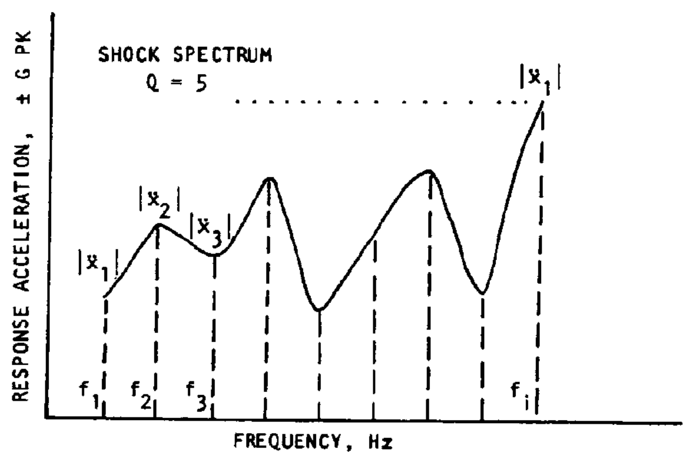
FIGURE 3-19. COMPARISON OF PROTOTYPE AND SCALED MODEL FOURIER MAGNITUDES OF ACCELERATION: 5TH FLOOR CENTER (Safford et al., 1977b)



(a) Mechanical reed gage to record peak motions at each reed natural frequency



(b) Mathematical model for digital calculation of shock spectrum, replacing mechanical method above



(c) Shock spectrum generated from (a) or (b) above

Figure 3-20. Methods of Measuring or Calculating Shock Spectra (Harris-Crede, 1976; BAC, 1977)

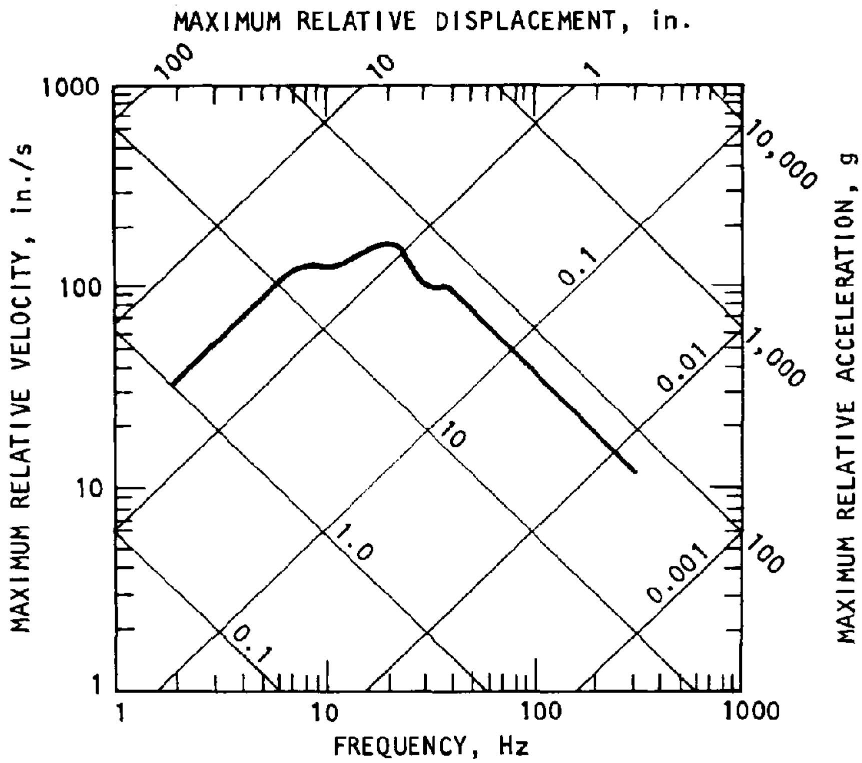


Figure 3-21. Example of Shock Spectrum (Ruzicka, 1967)

CHAPTER 4

EQUIPMENT FRAGILITY AND PERSONNEL TOLERANCES

4-1. Introduction.

a. Equipment that is inherently rugged enough to withstand the expected input shock motion can be hard mounted to the structure. All other equipment items, and perhaps personnel, will require shock isolation to reduce the shock input to a level the equipment/personnel can withstand. This chapter discusses how to determine, through measuring or estimating fragilities and tolerances, whether equipment and personnel can be expected to survive the anticipated facility environments. This information is also useful in determining the degree of shock isolation that may be required.

b. Standard methods for determining fragility are discussed and standard definitions are provided for fragility and fragility level, failure, malfunction, and damage. Also, three versions of fragility are defined: actual, acceptable, and modeled.

c. Fragility is a function of the dynamic character of the environment. Sensitivity of fragility to various environmental parameters such as application time is of obvious importance in evaluation. Parameter sensitivities of this nature are considered; those in particular are waveform simulation, transmission direction, level of assembly tested (part and subassembly), and low-cycle fatigue. Also important in evaluation is an awareness of the uncertainties in the environment and equipment. Some of the uncertainties are pointed out.

d. Depending on the stage of development, the type of information needed, and other constraints, different kinds of tests are used that relate to fragility estimation. These tests include diagnostic, fragility, qualification, production, and standard simulation tests. The purpose of these tests and how they may affect fragility estimates are described. Of particular importance are standard simulation tests, including shock tests, transient tests, and vibration tests. Such tests are considered in some detail.

e. It is not always possible to estimate fragility from tests. Perhaps the schedule does not allow time or appropriate test facilities are not available. In this case, fragility can often be estimated by comparing the equipment to another of similar construction for which fragility data are available. Some different weapon system equipments and commercial equipments are described for this purpose.

f. Personnel tolerance data are relatively scarce. However, some relevant data and estimates are presented. These data provide information on personnel tolerance to impact, on the effect of body position on tolerance, and the conditions under which injuries are most likely to occur. Additional information may be obtained on personnel tolerance and injuries from the U.S. Department of Transportation (National Highway Transportation and Safety Administration, the Federal Aviation Authority, and the National Transportation Safety Board) and from the U.S. Air Force Institute of Aerospace Medicine at Dayton, Ohio.

4-2. Determining fragility and tolerance levels.

a. Determining fragility and tolerance levels is often costly, difficult, and subject to uncertainty. The fragility levels of equipment items are usually estimated using one or more of several methods. The standard method of estimating fragility is through actual tests. However, this method may involve repeated testing under increasingly severe shock loading; variations in frequency content, waveform shapes, and duration of load; and changes in the direction in which loads are applied. Because of the time and expense involved in such procedures and the limited class of environments that make up the shock hazards for hardened facilities, such tests are seldom used to attempt a complete fragility evaluation. Nevertheless, tests will be discussed in some detail in this chapter, due to their fundamental importance in establishing and checking fragility levels.

b. Another method of estimating equipment fragility is by means of measurements and test results from other representative equipment and environments. The shock isolation designer should make use of this kind of data where possible. A large amount of fragility data for a variety of equipments has been collected in appendix A for this purpose. (A procedure for applying equipment fragility data to the evaluation of shock-isolation requirements is discussed in Bradshaw-Sonnenburg, 1979.)

c. For some items, the fragility level can be estimated analytically. This approach usually consists of constructing a mathematical model of the item made up of mass, spring, and damper elements. The model is examined with representative shock input motions and fragility levels are established from the calculated results.

d. Fragility determinations should take into account different categories of equipment. Prime mission equipment is normally of a special nature and is developed specifically to meet the requirements of the mission. Shock requirements are normally specified under these conditions and test demonstrations and stipulated. Facility support equipment, i.e., the bulk of protected subsystems equipment, is frequently available as off-the-shelf commercial equipment. Mission-critical commercial equipment must also be subjected to the facility shock environments and to environmental test verification similar to those on the specially designated prime mission equipment. Noncritical equipment needs only to structurally survive, or at the least not to interfere with mission operation due to structural or functional failure.

e. Knowledge of fragility or hardness levels permits an assessment of an elementary component's or complete system's vulnerability or probability of survival. Such assessments are made with respect to the expected dynamic environment. Fragility or hardness determinations also provide the following:

—Confirmation of system/component "weak links" early in the design cycle
—Identification of unanticipated weak links
—Guidelines for upgrading fragility level
—Minimization of need for redesign, replacement, and repair late in the development cycle or even later in field usage

f. Estimates of personnel tolerance to shock and vibration are based on a variety of tests and on accident summaries. However, experimental and analytical data regarding human shock tolerance are quite limited. Some of the more pertinent data, ranging from brief impacts to long-time accelerations, are summarized in the final section of this chapter.

4-3. Fragility Definitions.

a. IEEE Standard 344-1975 (IEEE, 1975) defined fragility and fragility level as follows:

Fragility. Susceptibility of equipment to malfunction as the result of structural or operational limitations, or both.

Fragility level. The highest level of input excitation, expressed as a function of input frequency, that an equipment can withstand and still perform the required functions.

b. Fragility of systems/components is usually categorized in terms of failure, malfunction, and damage. These terms indicate that fragility is a more comprehensive concept than initially expected: Its operational connotation extends the meaning well beyond structural breakage. However, even these fragility categories are subject to different interpretation in the different technical fields involved. The categories are defined as follows (Rountree-Safford, 1970):

(1) Failure is defined as an irreversible environment-induced inoperative condition, operation outside of tolerances, or change in operational state. "Irreversible" refers to the system/component remaining inoperative, out of tolerance, or in changed operational state after the environment is removed.

(2) Malfunction also represents an environment-induced inoperative condition, out-of-tolerance operation, or change in operational state. However, the process is reversible, i.e., the system/component returns to satisfactory operation upon removal of the environment.

(3) Damage has multiple meanings. It may be considered a mild form of failure (i.e., irreversible but borderline operation). It also represents permanent degradation in performance, reduction in hardness, or limitation of survivability. Damage also applies to system/component attributes that are unrelated to performance (e.g., deformation of missile support resulting in tilted but unimpaired launch).

c. At least three versions of fragility arise in practice. Which of these is pertinent depends on the stage of the design cycle or the state of knowledge of the system/component dynamic behavior. The three fragility versions are as follows:

(1) *Actual fragility.* Actual fragility is defined as the dynamic environment magnitude and its variation with frequency and time that is just sufficient to cause failure, malfunction, or damage (Safford-Inouye, 1957). Below this limit is the hardness region where systems/components are hard to the environment. Figure 4-1 (Himelblau et al., 1957) illustrates this concept in the form of a typical three-dimensional surface with uncertainty bands. The reduction in hardness along the time axis of figure 4-1 is characteristic of fatigue. Such a fragility usually falls in the failure or damage category. Malfunctions are normally not long-term-duration dependent. They are sensitive to short-term-pulse rise time and system/component critical period relationships. When fragility is independent of time, a two-dimensional plot of magnitude and frequency is appropriate, such as illustrated in figure 4-2 (Brust, 1961). This figure shows the actual fragility curves for each orthogonal axis of a motor-driven resolver.

(2) *Acceptable Fragility.* If a specification states that a system/component must pass a given test to be qualified, the conditions of this test define the acceptable fragility surface. Parameters to designate acceptable fragility surfaces vary. One abscissa of the magnitude-frequency-time surface is often left to interpretation by rescribing a magnitude-time history. Alternatively, shock spectra together with a required number of repetitions are often specified (Favour-LeBrun, 1969). The ordinates of the acceptable fragility surface can at best be equal to or less than the ordinates of the actual fragility surface. This relationship is shown in figure 4-3 (Himelblau et al., 1957). Unless the actual fragility is determined, it will not be known if the difference between the two surfaces is large or small.

Thus, the acceptable fragility surface establishes a specified minimum and nothing more.

(3) *Modeled fragility*. Early in the design cycle, the system operations analyst usually attempt to make a best estimate of the system/component vulnerability or ability to survive a dynamic environment. Tradeoffs are required between using new system/components, upgrading off-the-shelf versions, or protecting off-the-shelf versions to a higher degree. At this stage of the design cycle, prototypes are not available so the analyst must rely on mathematical or empirically derived models to estimate fragility. This is referred to as modeled fragility.

4-4. Parameter sensitivities in fragility evaluation.

a. General. Fragility evaluation requires that the sensitivities of various parameters be considered. Cri-

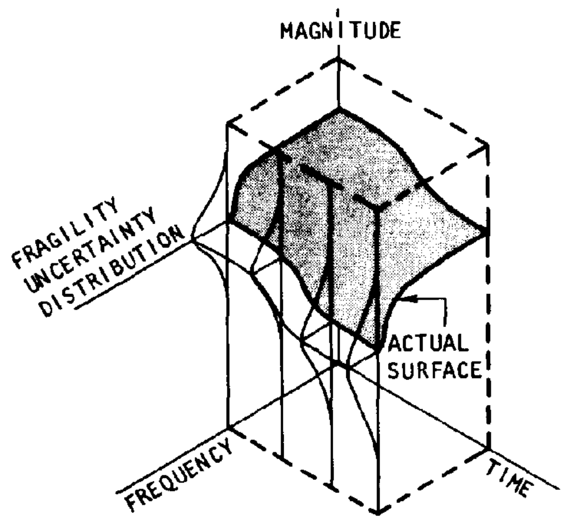


FIGURE 4-1. ACTUAL FRAGILITY SURFACE (Himelblau et al., 1957; reprinted with permission copyright 1957 Soc. Auto. Eng. Inc.

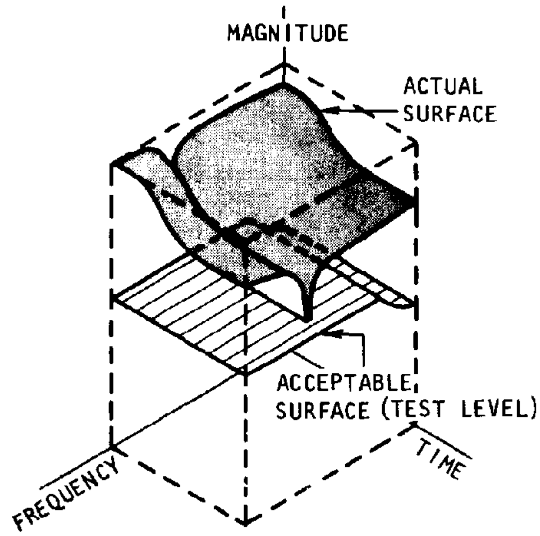


Figure 4-3. COMPARISON OF ACTUAL WITH ACCEPTABLE FRAGILITY SURFACE (Himeblau et al., 1957; reprinted with permission copyright 1957 Soc. Auto. Eng. Inc.

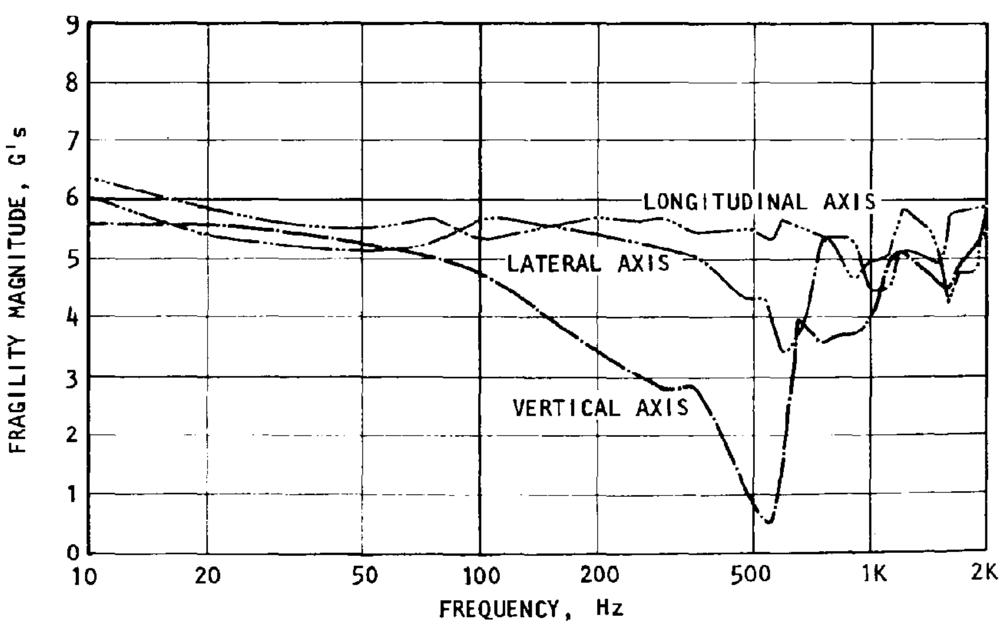


FIGURE 4-2. SINUSOIDAL FRAGILITY CURVES FOR A MOTOR-DRIVEN RESOLVER (Brust, 1961; reprinted with permission copyright 1961 Soc. Auto. Eng. Inc.

teria for what constitutes failure, malfunction, or damage are determined from the operating function of the system/component under evaluation. Nevertheless, it must be recognized that, in general, fragility or hardness is a function of the dynamic character of the environment. Obviously, a constant 1 g application will have a different effect from 1 g for 1 msec or 1 g slow sine wave sweep applied at a set frequency. Directional sensitivity is another variable to be strongly considered. For example, in figure 4-2, note that except for the regions 17 to 60 Hz and above 1100 Hz, the vertical axis has the lowest and therefore the controlling fragility level.

b. Specification of environment.

(1) Since actual fragility is dependent on dynamic environment, it would be desirable to use the exact environment pertaining to system/component application in the evaluation. But this requirement is clearly impractical. Either the exact environment is unknown or it cannot be accurately simulated in the fragility evaluation, or both. Furthermore, "exact" is a misnomer, since the intended application is generally in a statistically described environment.

(2) Experience and engineering judgment are often used to bridge the uncertainties between simulated and realistic environments. The need for an exact specification of environment is reduced by the existence of standardized test specifications. Also, instead of an exact specification of dynamic environment, general categories of environment are used. Examples of such categories of environment are shock, transients, and vibrations. These categories recognize the effects of various loadings on system/components.

c. Waveform simulation. Transient shock environments transmitted by nuclear protective structures to weapon systems or equipment-mounting locations are characterized by an amplitude-wave shape, time duration, and multifrequency content. The transient motion at any point in the structure is a time-varying vector in spherical space, which is normally represented as motion-time histories in three orthogonal axes. Fragility tests or environmental qualification tests should at least approximate the generic wave forms expected at the attachment or mounting locations of the equipment or system (Rountree-Safford, 1970). If the external input is impulsive in character, then simple pulse shapes can be considered where amplitude, duration, frequency content, and shape are controlled. For the cases of random (shaped for the transmission path), sine, or multifrequency inputs, constraints similar to the above apply.

d. Transmission direction.

(1) Transmitted environments occur simultaneously in all directions. The direction of test motion should ideally be in all three principal axes simultaneously. At the present time, biaxial test facilities are limited and triaxial facilities are just being introduced.

(2) If the equipment being tested can be shown to respond independently in each of the three orthogonal axes, a single-axis test is appropriate. The cross coupling must be very low; otherwise, conservative qualification test levels must be used. For evaluating fragility, an uncertainty must be added to the normal statistical variation expected.

(3) Biaxial testing normally is performed using the vertical and one horizontal axis, followed by a second test using the vertical and the other horizontal axis. Input motion is the predicted acceleration-time histories (vertical and horizontal) for each axis, or the prescribed qualification test environment.

e. Level of assembly: part/component, subassembly, assembly, subsystem, or system. A general rule for test is the higher the level of assembly, the better for fragility evaluation or qualification. Size and weight normally dictate testing at the unit, subsystem, or assembly level consistent with test machine capacity. At the lower assembly levels, special instrumentation to monitor functional operation is often required. The local input threat environment must also be determined if the test article is mounted on an equipment or operating unit.

f. Low cycle fatigue.

(1) Equipment and weapon systems housed in protective structure are expected to withstand one or more attacks where the shock and vibration loading is of a transient nature with the loading duration not exceeding a few seconds, and possibly only milliseconds. Additionally, in-place equipment and weapon systems may have an environmental shock and vibration test history from qualification tests and quality control production vibration tests. Low-cycle material fatigue must be considered where the transmitted transient environment induces high amplitude stresses. This is especially required where equipment used for qualification tests and quality control production shock and vibration tests will be subsequently installed in a weapon system.

(2) Past histories of testing and operations on a wide variety of equipment have demonstrated for the most part that malfunctions of equipment and weapon systems are not time dependent or at the most are dependent only in special cases.

(3) Cumulative damage theory may be applied to a load-carrying structure of equipment with reasonable confidence (Fackler, 1972). The more conservative Palmgren-Miner theory is described here. A damage index D is defined as follows:

$$D = \sum_i \frac{n_i}{N_i} \qquad (4\text{-}1)$$

where

n_i = Number of cycles actually occurring at stress amplitude σ_i

N_i = Number of cycles necessary to cause failure in fatigue at constant stress amplitude σ_i

The damage index D is a number having the value $0 < D \leqslant 1$, and failure is considered to occur when D = 1.

(4) A relationship is postulated between response stress levels and cycles to failure that can be expressed as $\sigma^\alpha N$ = constant. Then if σ_{max} is the stress level that will cause failure after one cycle:

$$\sigma^\alpha N = \sigma_{max}^\alpha \qquad (4\text{-}2)$$

or

$$\frac{1}{N_i} = \left(\frac{\sigma_i}{\sigma_{max}}\right)^\alpha \qquad (4\text{-}3)$$

Test data (Safford, 1974) may be introduced from the various test levels in the following example for one of three axes. For item j

$$\frac{1}{3}D^j = \left(\frac{\sigma_o}{\sigma_{max}}\right)^\alpha \{\underset{\substack{200\%\\ \text{level}\\ \text{test}}}{1} + \underset{\substack{150\%\\ \text{level}\\ \text{test}}}{4(0.75)^\alpha} + \underset{\substack{100\%\\ \text{level}\\ \text{test}}}{9(0.5)^\alpha} + \underset{\substack{75\%\\ \text{level}\\ \text{test}}}{15(0.375)^\alpha} + \underset{\substack{50\%\\ \text{level}\\ \text{test}}}{23(0.25)^\alpha}\} \qquad (4\text{-}4)$$

where 1, 4, 9, 15, and 23 are the numbers of cycles, respectively, at the levels shown. On the basis of extensive tests with various types of equipment components (Safford, 1974), the slope parameter α is ≈ 9.

(5) Evaluating the above equation with $\alpha = 9$ for one axis yields

$$\frac{1}{3}D^j = \left(\frac{\sigma_o}{\sigma_{max}}\right)^\alpha [1 + 0.30 + 0.018 + 0.002 + 0.00009] \qquad (4\text{-}5)$$

By comparing the relative contribution of the different test levels to the cumulative damage index, it is obvious that the 200 percent and 150 percent are the significant ones.

(6) The damage index for specimen (j) can be related to the test level by

$$\frac{\sigma_{max}}{\sigma_o} = 1.16\, D^{-1/9} \qquad (4\text{-}6)$$

As a result of a fragility evaluation of the results of the shock test program, it is estimated that the value of the cumulative damage index D is ≈ 0.1. This results in

$$\frac{\sigma_{max}}{\sigma_o} \approx 1.50 \qquad (4\text{-}7)$$

This value indicates that the test levels could have been scaled upward by a factor of ≈ 1.5 times the maximum or 200 percent test level before significant damage would have occurred.

(7) Consequently, based on the experimental results of the shock test, it can be predicted that the tested equipment (main frame structure) could have withstood approximately 300 percent test levels before suffering major damage. A representative sample of specimen response is the histogram of peak responses in figure 4-4. The maximum level of excitation was 200 percent for a rack-mounted communication data set.

g. Stochastic effects and uncertainties. The evaluation of fragility must include consideration of some stochastic effects and uncertainties. These include uncertainties both as regards environment and equipment.

(1) Environment. The operational environment is never known precisely, and its relation to the test environment is uncertain. Qualification tests may not be representative of the expected input environment. There may be cross-axes coupling due to a multiaxes environment, which includes rotational inputs, although the equipment has been qualified by single-axis tests.

(2) Equipment quality control. Fragility tests on identical units would be expected to reveal a statistical distribution such as illustrated in figure 4-1. Qualification tests such as shown in figure 4-3 would exhibit no failures or malfunctions if the test level were below the statistical scatter of the unit's fragility. Qualification test levels that lie within the fragility scatter, however, would require design modification for hardness upgrade. Implicit in the statistical scatter is the requirement that the production process, including material and purchased parts/components, is under quality control.

(3) Test results. Tables 4-1 and 4-2 list examples of defects found during qualification and fragility tests on commercial telephone equipment. The 100 percent test level corresponded to the input level of figure 4-10a and b. The 200 percent test level was twice the amplitude of these figures (see also appendix A-3). These data have applicability for isolated systems in protective structures where the threat is of

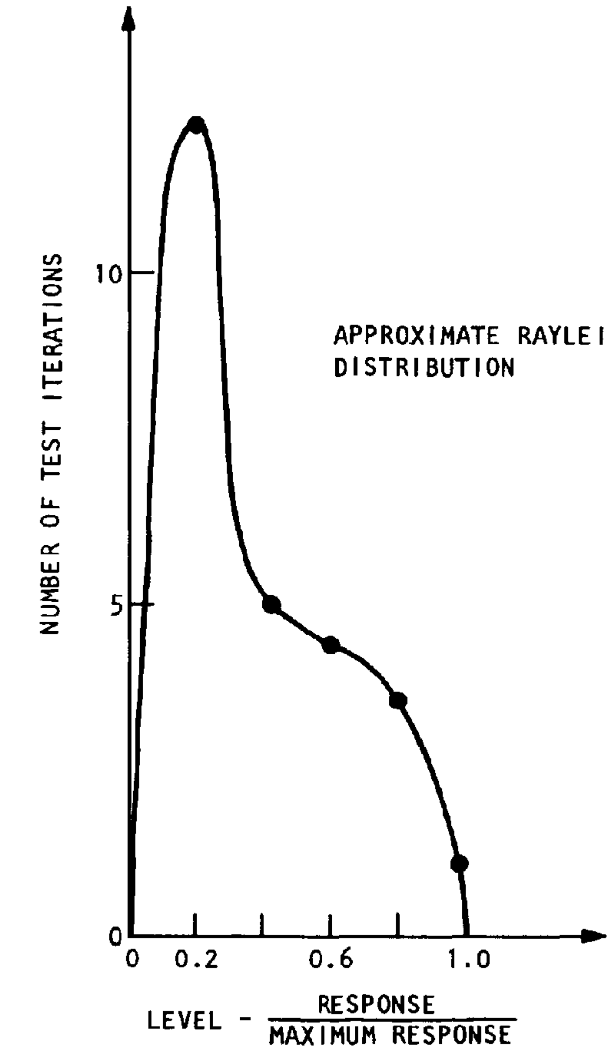


Figure 4-4. Communications Data Set, Histogram of Responses (Safford, 1974)

comparable magnitude, frequency content, and duration. Equipment structural characteristics may be extracted from the above tests in the form of transmissibility, transfer functions, and modal characteristics of shapes, frequencies, and damping.

4-5. Testing.

a. General. Tests used in shock isolation system design include diagnostic, fragility and qualification, and production tests.

b. Diagnostic tests.

(1) Diagnostic tests are useful in exploring the response characteristics of an equipment structure and the functional operation of the equipment. The primary purpose of diagnostic tests is the observation of failures and malfunctions that may be related to excitation frequency, amplitude, test level, or impulse, depending on the test employed. This permits identification of weak links and the rational hardening/upgrade of equipment.

(2) The tests normally include input excitations in the following forms, without attempting to simulate eventual shock or vibration environments:

—Slow sine sweeps and resonance dwell
—Random vibration
—Rapid sine sweeps (chirp)
—Shock tests
—Sine beat tests

c. Fragility and qualification tests. Unlike diagnostic tests, fragility and qualification tests attempt to simulate the environment, at least as to waveform. In fragility tests, the test level is raised until failure or malfunction occurs, establishing the fragility level of the equipment. In qualification tests, a test level is selected that is based on the threat environment plus safety factors to allow for uncertainties. If the equipment passes the test, this establishes a lower bound for equipment fragility. The equipment is not tested to

TABLE 4-1. QUALIFICATION (100 percent) LEVEL TEST SUMMARY (Safford-Tuttle, 1974)
(See figures 4-10a and 4-10b; also appendix A-2)

Equipment	Failure/Malfunction/Degradation: Operational Effect	Failure/Malfunction/Degradation: Mode	Failure/Malfunction/Degradation: Mechanism	Failure/Malfunction/Degradation: Cause	Test Level Intensity, Percent	Corrective Action	Results
111A Power Equipment	Degradation (reduced capacity)	Fuse holder fell off	Low spring tension on connectors	Design	100	Fuse panel restrained	Hardened to 100%+
	None	Voltmeter panel fell off	Low friction restraint	Design	100	Panel restraints or latches	Hardened to 100%+
		Doors swing open		Design	50	Door latches	Hardened to 100%+
Inverter	None	None	--	--	100	--	Passed qualification level 100%
Battery Rack	None	None	--	--	100	--	Passed qualification level 100%
Crypto	None	None	--	--	100	--	Passed qualification level 100%
203/303 Data Equipment	None	None	--	--	100	--	Passed qualification level 100%
46A Multiplex Unit No. 1	Degradation (reduced capacity)	4 kHz Generator module disengaged	Low spring tension on connector	Design	50	Module restraint brackets	Hardened to 100%+
46A Multiplex Unit No. 2	Degradation (High error counts in data stream)	Intermittent shorts	Loose wire	Quality control	100	Discovered and corrected at 200% test	See fragility test (Table 4-2)
46C Terminal Equipment	Malfunction (error counts and interruptions, 1 kHz tone)	Line input unit, loose jumper assy	Low spring tension and friction, connector jumper assy	Quality control or design	50	Split end connectors spring force increased	Hardened to 100%+
46C Dependent Terminal	Degradation (minor effect on isolator)	One friction damper failed to operate	Isolator damper guide pin fell out	Quality control	100	Reassembled damper	Passed qualification level 100%

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TABLE 4-2. FRAGILITY (200 percent) LEVEL TEST SUMMARY (Safford-Tuttle, 1974)
(Two times magnitude level of figures 4-10a and 4-10b; see also appendix A-2)

Equipment	Failure/Malfunction/Degradation: Operational Effect	Mode	Mechanism	Cause	Test Level Intensity, Percent	Corrective Action	Results
111A Power Equipment	Degradation (reduced capacity)	Fuse holder disengaged	Low spring tension and friction of connector	Design	200	Fuse panel restrained	Hardened to 200% +
Inverter	None	1/4" bolt sheared	Cumulative fatigue	Test dependent	150	Bolt replaced	Passed fragility level 200%
	Failure	Short circuit	Loose bolt on capacitor	Quality control	200	Capacitor remounted	Passed fragility level 200%
Battery Rack	None	None	--	--	• 200 vertical • 150 X- and Y-axis	--	• Passed fragility level 200% vertical • Passed level 150% horizontal axes
Crypto	None	None	--	--	--	--	200% level
203/303 Data Equipment	Failure (data signals lost)	Power supply fell out of rack	Weak mounting bracket	Design	X-axis 200 and Y-axis 200	Support bracket installed	Hardened to 200% +
46A Multiplex Unit No. 1	Failure (data signals lost)	Plug-in modules disconnected	Low spring tension on connector	Design	150	Restraint bars added	Hardened to 200% +
	Degradation (slight signal interruption)	Relay contact chatter	Armature dynamics	Design	150	None	Passed fragility level 200%
46A Multiplex Unit No. 2	Failure (data signals lost and 1 kHz tone interruptions)	Fuse operated, lost power to capacitor bank	Extraneous loose wire, electrical short	Quality control	200	Removed loose wire	Passed fragility level 200%
46C Terminal Equipment	None	--	--	--	200	See qualification test	Passed fragility level 200%
46C Dependent Repeater	None	--	--	--			• 150% X and Y axis • 200% vertical axis

U.S. Army Corps of Engineers

failure. Because of cost and other factors, fragility tests are often performed at the component level, while qualification tests are generally performed at the assembly or system level.

d. Production tests.

(1) After manufacture, production or quality control tests may be performed to reveal weaknesses or defects due to errors or excessive variability in manufacture of the equipment. Production tests may be applied to each equipment delivered or installed in a weapons system. Type of tests, test levels, number of tests, and their duration at the maximum should not exceed the qualification or fragility levels of the prototypes and normally are at lesser levels sufficient to assure mission critical functions during and after an attack.

(2) Selection of quality control/production tests may be somewhat arbitrary and may not necessarily attempt to duplicate environmental simulation of a qualification or fragility test. Vibration and shock machines may be used at the place of manufacture, or tests may be performed by a testing laboratory. Equipment shipped by common carrier is subjected to a variety of vibration and shocks that can best be summarized in a statistical sense (NRL, 1955). Other types of testing that are particularly adapted for large systems or for use after installation in the protective structures may be employed. These tests can include shock and vibration tests applied directly to load-carrying elements of the equipment.

4-6. Standard simulation tests.

a. General. The preferred method of determining the fragility of a system is to test it with all components intact. A realistic dynamic environment is used while operations or functional parameters are monitored. Unfortunately, a complete system is often not available until late in the design cycle, or there may be economical and technical limitations. Instead of the preferred method, therefore, tests are usually conducted at a component level. Failure, malfunction, or damage criteria pertaining to a system specification must then be interpreted for each component. System fragility requirements are allocated to components in the form of a fragility budget. Correspondingly, system fragility must be estimated from component fragilities. The accuracy of these estimates, both at the component and system levels, is primarily limited by testing constraints.

b. Testing constraints.

(1) Testing constraints apply to fragility determination as well as to other testing (e.g., qualification). The best test conditions must be determined by interpretation and experience, supported by analysis. Then test results must be derated or enhanced for field applications. As an aid in this task, force measurements should be obtained in addition to motion measurements at the test item/test fixture interface (Ballard, 1969; Painter-Parry, 1966). Such measurements are necessary to adapt the fragility curves to the field installation.

(2) Test specifications generally imply an unrealistic "irresistible input motion" on the part of the text fixture. The major testing constraints are:

—Restriction to single-axis testing
—Limitation of size or input of the test equipment
—Lack of identification of the test fixture impedance
—Lack of repeatability (i.e., lab-to-lab or day-to-day)
—Unavailability of sufficient numbers of test specimens
—Inaccessibility of operational or functional measurements
—Inability to generate combined environments

Standard simulation tests often used for shock, transient, and vibration environments are described next.

c. Shock tests.

(1) Shock tests are a special case of a transient function, described in the next subsection. Shock tests are simple pulse tests. The most popular pulse shape used to date is the terminal peak sawtooth pulse (McWhirter et al., 1967). Other pulses may be selected, such as the rectangular pulse that has recently had a surge of interest (Goff-Pierce, 1969) in the packaging field.

(2) Overall control of the frequency content of the shock pulse is dictated by shock spectra. Any of the pulses on the right in figure 4-5 may be selected for the test, provided it generates the appropriate level shock spectrum. Levels begin at a sufficiently low value to be under the fragility level. The beginning level is identified in the figure as level 1. The level is subsequently increased to level N, which corresponds to a failure, malfunction, or damage (Vigness, 1964; Schell, 1964; Palmisano-Kaplan, 1966; and Jacobsen-Ayre, 1958).

(3) Both drop tower shock machines and large vibration shakers are usually employed for these types of shock tests. The carriage of the drop shock machine is arrested in its fall by a lead or a lead-base pellet that forms the pulse. Vibration machines are programmed to produce the pulse waveform and are normally limited to a maximum 1-in. stroke. Other types of shock machines have been developed and are classed, for example, as resilient rebound, nonresilient, hydraulic, or compressed gas (Vigness, 1964). Specifications covering shock tests are covered by Method 516 of MIL-STD-810 and the *General Specification for Environmental Testing, Aeronautical and Associated*

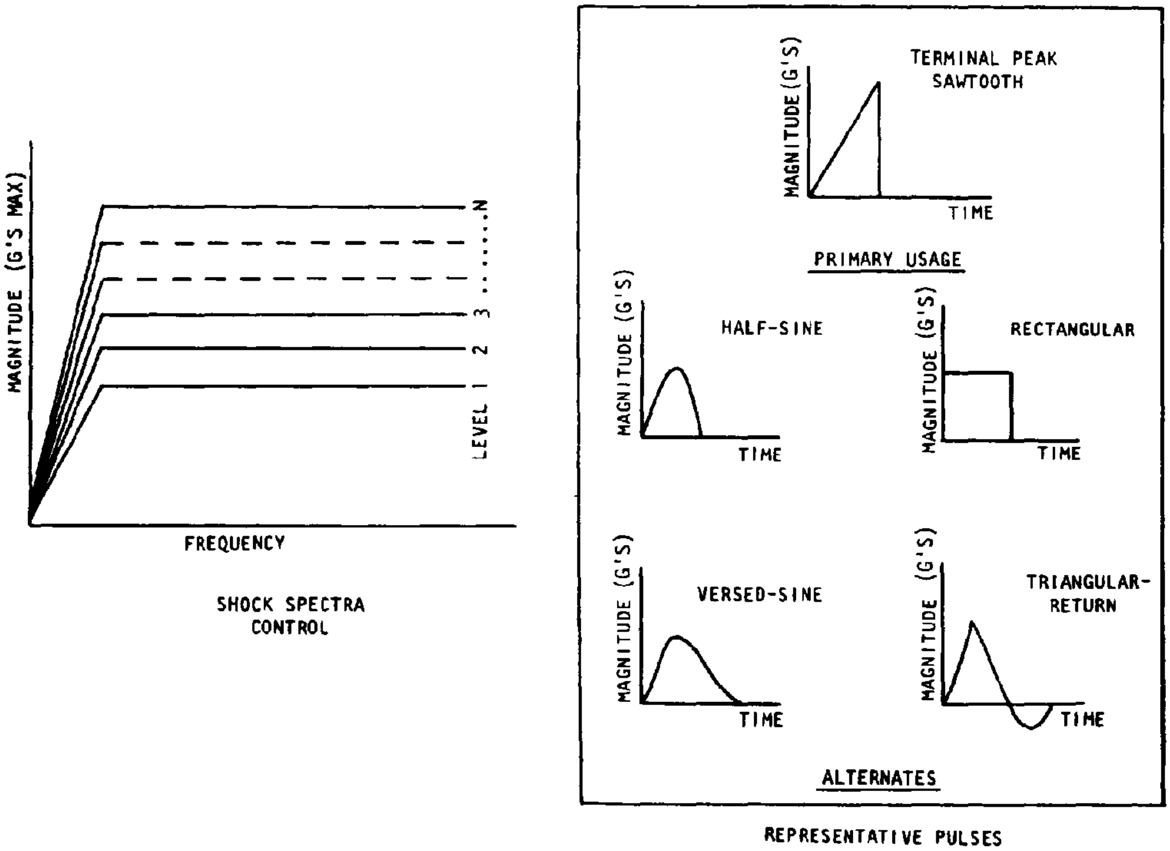


Figure 4-5. Standard Shock Fragility Test (Rountree-Safford, 1970)

Equipment, MIL-E-5272.

(4) Damage boundaries (fragility) for equipment have been developed using acceleration shock pulses as input (Goff-Pierce, 1969). Variations of peak acceleration and pulse durations produces limiting velocity changes to generate curves as shown in figure 4-6. Frequency dependence is not evident in this procedure, although frequency spectrum levels may be derived by Fourier transformation of the pulses. In view of the comments in Schell (1969), this procedure can be considered approximate, being over or under conservative. This over or under conservatism can be reduced to a considerable extent by use of a terminal peak sawtooth pulse form.

(5) An additional class of shock tests arises from the U.S. Navy Requirements for Shipboard Machinery, Equipment, and Systems, "High Impact Shock Tests," MIL-S-901C. This test procedure is defined as a rough approximation to battle damage environments. Four classes of tests facilities are considered:

Shock-testing machine for lightweight equipment (to 400 lb), figure 4-7; Shock-testing machine for medium-weight equipment (250 to 7,400 lb), figure 4-8; and two classes of Shock-testing of heavy equipment. The heavy equipment is mounted on floating shock barges, where the forcing function is initiated by underwater explosives. The nature of the motion induced on these shock barges is transient. The lightweight and medium-weight navy shock machines employ an impacting (falling weight) hammer. The test criterion for these machines is the response spectra of the equipment. Input acceleration-time motions at the equipment mounting points are shaped by the equipment input impedance, fixturing, motion-limit stops, or in general, the impedance path between hammer impact point and equipment. As such, the shock waveform input is not in the simple form shown in figure 4-5.

(6) For the standard shock machines and for various loadings and modes of operation, shock spectra have been reported that can be used to determine acceleration response capability. Figure 4-9 shows typical shock spectra for several shock testing machines. Additional information on U.S. Navy shock machines is contained in NHA (1963) and Vigness (1961).

d. Transient tests.

(1) General

(a) While shock test inputs have durations in milliseconds, transient tests have durations of up to one or two seconds. Acceleration-time history inputs are synthesized from decaying sinusoids, by sine-beats, and shaped random motion (IEEE, 1975; Safford,

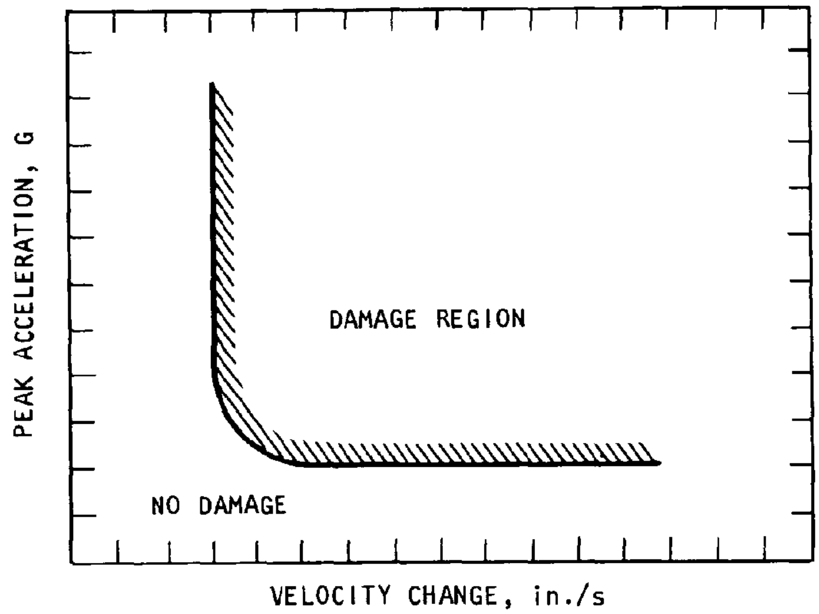


Figure 4-6. Damage (Fragility) Boundaries for Equipment from Pulse Tests (Golf-Pierce, 1969)

1974; and Yang-Saffell, 1972) as approximations of transmitted waveforms and durations to match criteria shock spectra.

(b) Vibrations machines—electrohydraulics or electrodynamic—are used for this type of test. The generation of a time history to match a shock spectrum is shown in figures 4-10a and 4-10b (Safford, 1974). The motion imparted to the test article is given at 2 percent and 10 percent damping of the shock spectra. Ratio of the amplitudes of these two curves provides information on the number of oscillations at each frequency (Crum-Grant, 1970).

(2) Transient functions. Transient functions are defined as amplitude time histories that are nonzero over a specified time interval (Ballard et al., 1969; and Morse, 1964). This usage of transient includes the wide range of motions from the analytic pulse to the onset of stationary processes. The significance of this range is portrayed in figure 4-11, which is a graph of amplification factor vs. quality factor (reciprocal of two times the damping ratio) for a simple mechanical oscillator. The graph illustrates three pertinent relationships among shock, transients, and vibrations:

—Steady-state vibration has a slope of unity
—Shock has a slope of approximately zero
—Transients have slopes between the limits of unity and zero

Figure 4-11 emphasizes the third relationship, i.e., a wide variety of dynamic loadings fall into the transient category. This category also accounts for the number of cycles that occur at various frequencies in the transient spectrum.

(3) Rapid sine sweep. One transient technique consists of a rapid sine sweep over the freqency range of interest. Predicted or measured shock spectra, based on two different quality factors (normally Q = 5 and Q = 25) are used to describe the environment. The ratio of these two shock spectra (i.e., Q = 25/5 = 5) governs the number of oscillations occuring at each frequency. Sweep time over the frequency range is also controlled by the number of oscillations at each frequency and the bandwidth. Sweep time usually varies from ½ sec to 3 sec duration. Sweeps are made both up and down the frequency scale. Figure 4-12 illustrates this type of transient test. Levels of the shock spectrum for each quality factor are increased to the point of fragility without changing the shock spectra ratio. This test covers all frequencies of concern that may be transmitted to the equipment, but in a sequential manner. Sweep time allows controlled buildup of resonances in the equipment under test. The relation of rapid sine sweep to actual environment is not well established, and it application implies conservatively high test loads. This type of test was used on weapon system equipment of the Minuteman, replacing the terminal peak sawtooth pulse tests described in the previous section.

(4) Sine beat and decaying sine tests. Sine beat and decaying sine tests are comparable to rapid sine sweeps in controlling the number of oscillations at each test frequency and consequently the time of environmental exposure. As these tests are applied a single frequency at a time, they may be viewed as diagnostic. The amplitude and number of cycles are chosen to match criteria or obtain fragility shock spectra. The relation of sine beat and decaying sine tests to actual en-

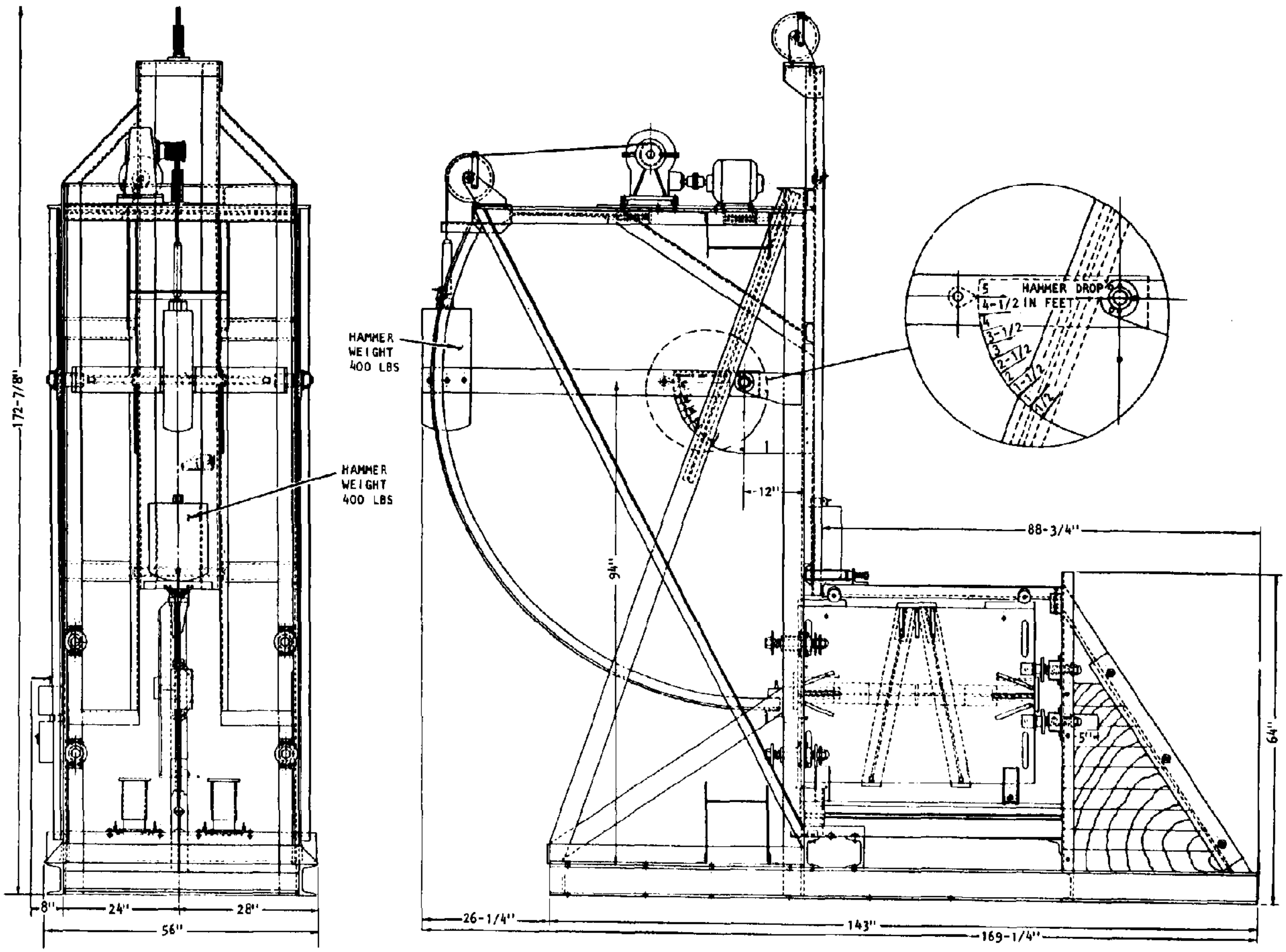


Figure 4-7. Shock-Testing Machine for Lightweight Equipment (Navy, 1963)

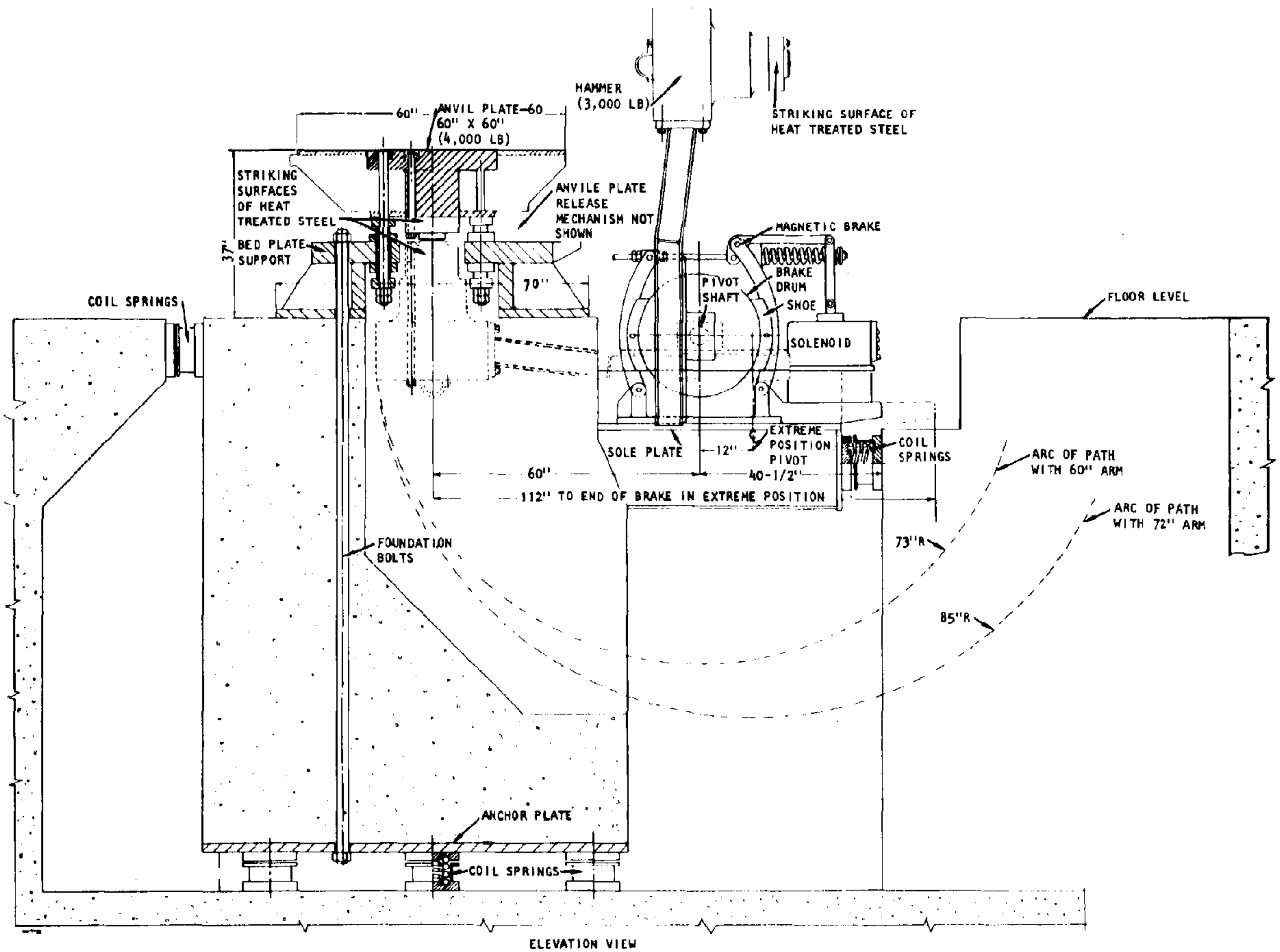


Figure 4-8. Shock-Testing Machine for Medium-Weight Equipment (Navy, 1963)

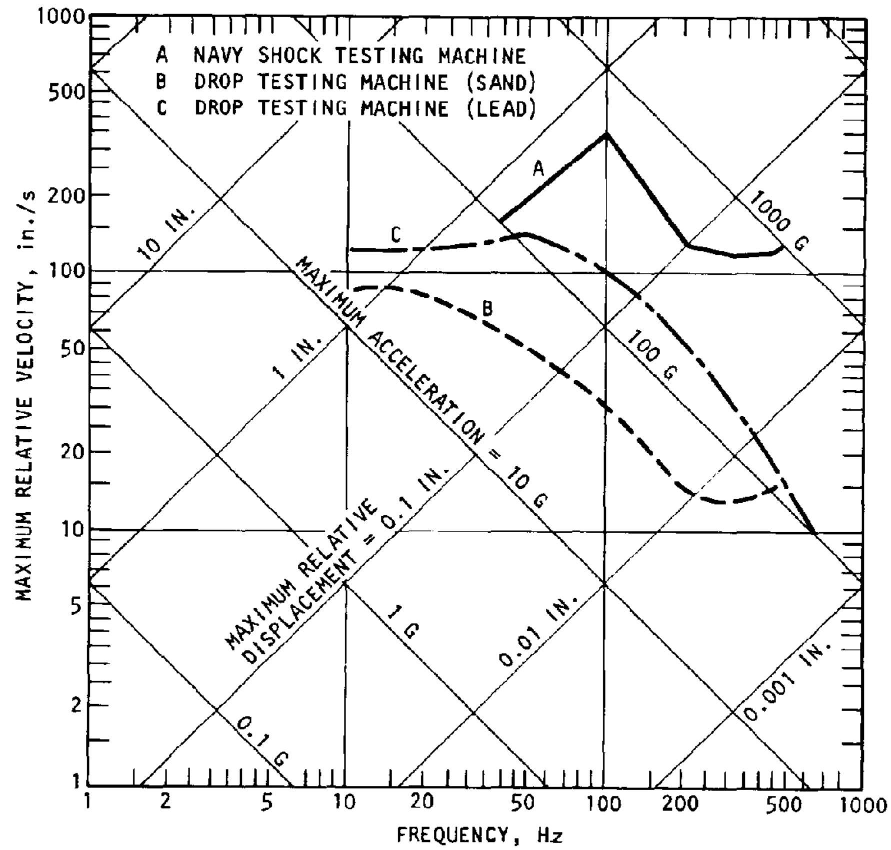


Figure 4-9. Typical Shock Spectra (Ballard et al., 1969)

vironments is not established; their application implies conservatively high test levels. Figure 4-13 for sine beat and figure 4-14 for decaying sine test illustrate the input test wave for each test frequency (IEEE, 1975).

(5) Multifrequency tests. Multifrequency transient tests are particularly apt for producing simultaneous response from all modes of a multidegree-of-freedom equipment or system under test. Multiple-frequency testing provides a closer simulation to a threat environment and minimizes the requirement to introduce a higher degree of test level conservatism. Three types of tests may be considered:

—Predicted time history
—Random motion
—Complex wave

(a) A series of predicted time histories produced by the threat environment at the mounting location of equipment provides the best simulation for dynamic loads. These inputs should range from the minimum or lower uncertainty level to the maximum uncertainty for qualification tests. Higher levels may also be employed for fragility tests.

(b) Random motion tests may be used, particularly where predicted time histories are not available. The random motion is generated separately for each test and is shaped to an estimated criterion or test shock spectrum with random time history duration approximating the expected transient times. Test levels range from minimum to maximum uncertainty levels. For fragility, higher levels would be employed.

(c) An application of a random approximation is shown in figures 4-15 and 4-16 (Safford, 1974). A remote communications repeater was enclosed by a shallow-buried protective shelter. Free-field ground motions induced by airblast loading of the ground surface generated an air-induced portion and a late-time Rayleigh wave motion, as shown in figure 4-16. Other cases included an outruning wave. Analytic waveforms of this type are subject to parameter variation to match the shock spectrum criterion given in figure 4-15, but for the systematic region only, as also noted in the figure. The region from 30 Hz to 1000 Hz in Figure 4-15 is defined as the structure/media interaction and represents the vibration induced by the ground shock. Free-field coupling with the structure over the entire time history is conservatively taken as unity. As the transient vibration response of the protective structure was only ascertained by the shock spectrum, random motion for each test level was superimposed on the air-induced portion of the free-field wave as given in figure 4-16. The random motion was shaped by

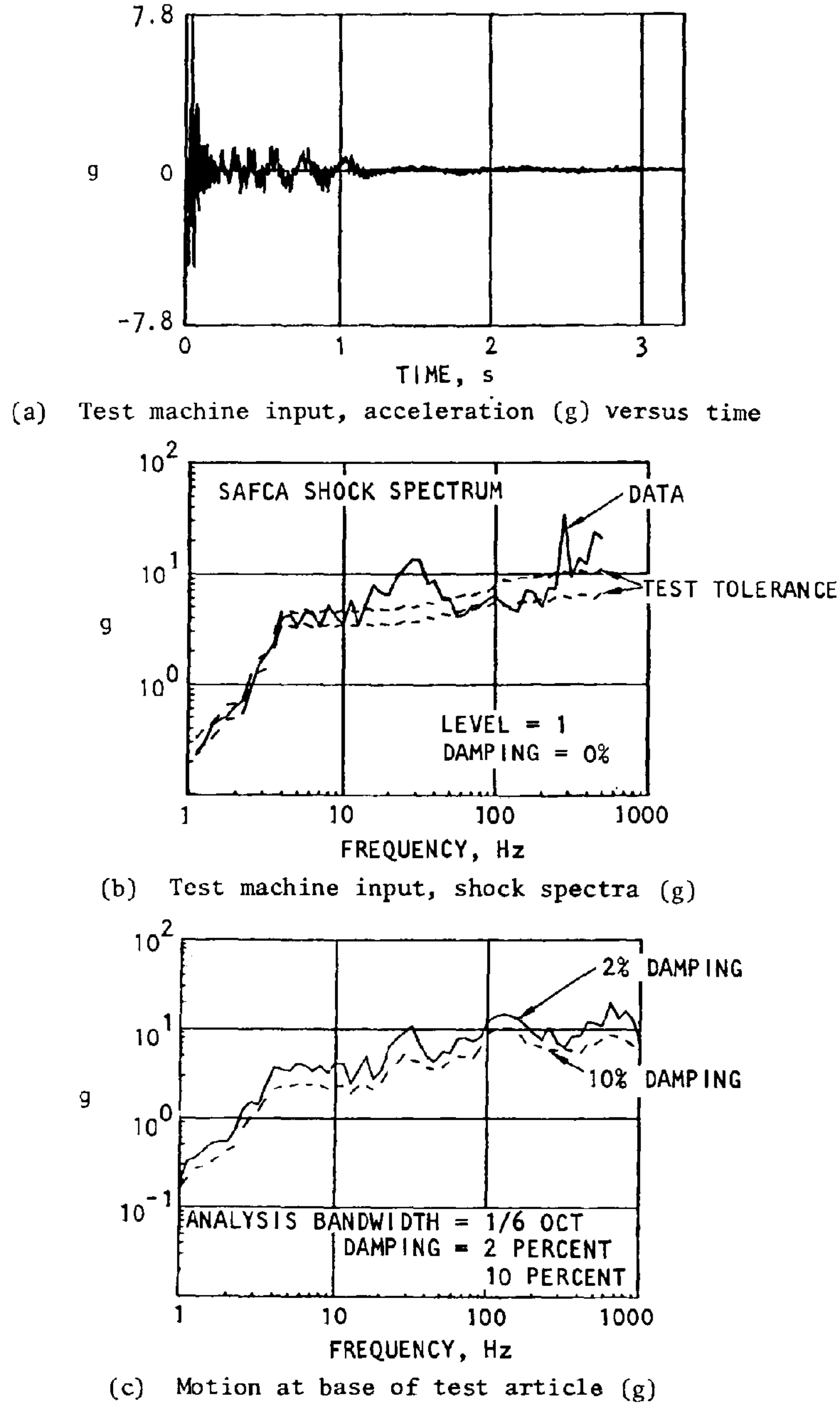


Figure 4-10. Typical Horizontal Test Machine Input and Motion at Base of Test Article (46 A Unit 1 Multiplex) (Safford, 1974)

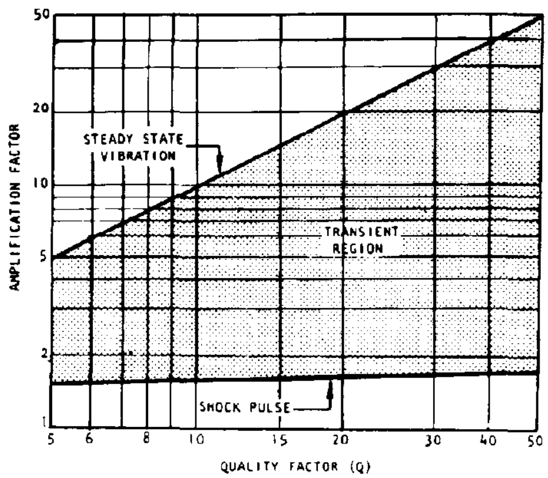


FIGURE 4-11. AMPLIFICATION VS. QUALITY FACTOR FOR MECHANICAL OSCILLATOR (Rountree-Safford, 1970)

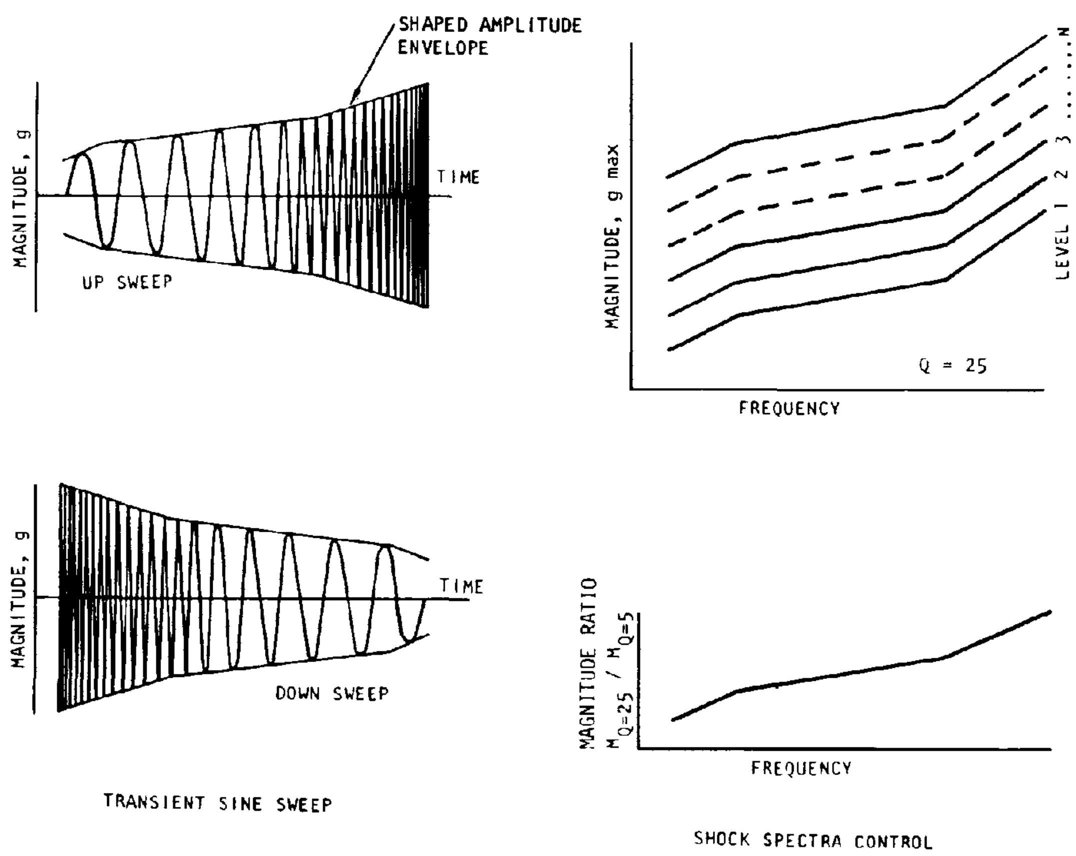
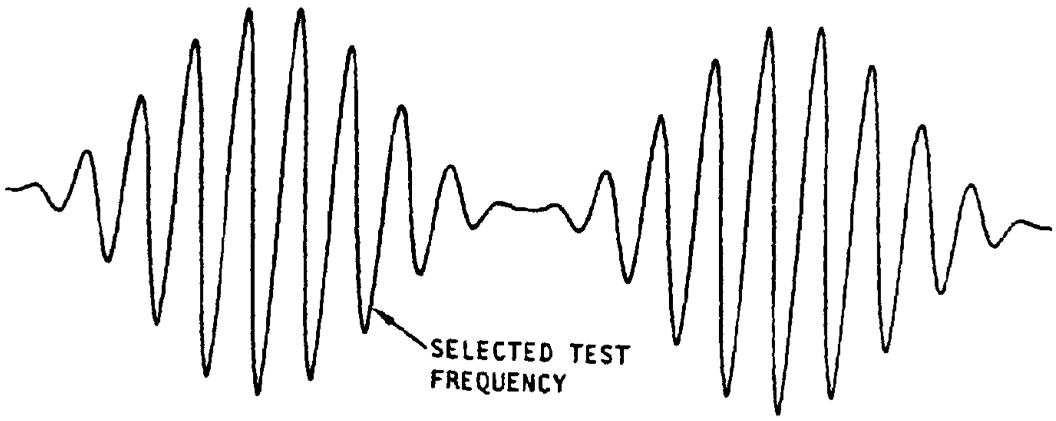


FIGURE 4-12. TRANSIENT FRAGILITY TEST, RAPID SINE SWEEP (Rountree-Safford, 1970)



Corps of Engineers

FIGURE 4-13. SINE BEAT TEST

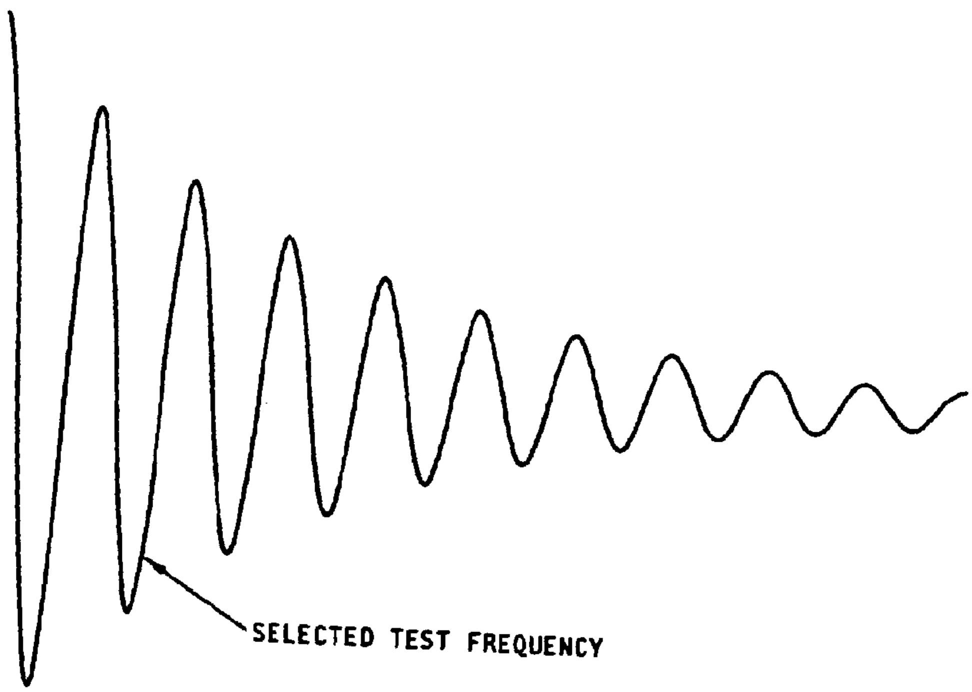


FIGURE 4-14. DECAYING SINE TEST

U. S. Army Corps of Engineers

the time duration of the air-induced wave, the shock spectrum levels, and frequency bandwidth. The time history of figure 4-16 was programmed to drive a vibration shaker for tests of the communications repeater. Because of the structural response uncertainty, different random motions were used for different test levels and test axes.

(d) Complex wave tests are synthesized time history motions to match shock spectra. The shock spectra are criteria or estimated environments for the equipment and normally range from the lower to upper uncertainty bounds for qualification, and higher for fragility level testing. One method of generation is by summing a group of decaying sinusoids. The frequencies of the component signals should be spaced at one-third octave or narrower frequency intervals to cover the the range of the specified shock spectrum. The detailed procedure is covered in IEEE (1975). Another method of generation is by summing a group of sine beats fig. 4-13) to match the shock spectra (Yang-Saffell, 1972). A criterion shock spectrum is shown in figure 4-17; a sine-beat synthesized waveform that will match the shock spectrum is given in figure 4-18.

(6) Shock barge tests. Shock barge tests are covered by U.S. Navy Specifications MIL-S-901 for a floating shock-test platform for equipment weighing up to 60,000 lb (fig. 4-19). Subsequent to the issue of this specification, a large floating shock test platform was developed and placed in operation (Schrader, 1974; Clements, 1974). This large test system handles equipment up to 400,000 lb. Transient time histories are dependent upon the mounting location on the barge, equipment weight, explosive charge weight, charge distance, charge depth, charge/barge orientation, and water depth. Typical velocity waveforms obtained from the barge tests are presented in figure 4-20 (Clements, 1974).

e. Vibration tests. Vibration tests are composed of both sine and random tests. Both have variable magnitudes as a function of frequency, and test times are in hours or fractions thereof.

(1) Sine wave testing consists of slow sine sweeps over the frequency bandwidth and includes resonance dwells. Sine wave testing has been historically used for qualification of aircraft equipment. The Air Force position is that equipment passing the sine wave tests has performed well in service operating environments. This claim results from the very conservative nature of the tests rather than their simulation of the environment, which for jet aircraft and missiles is random. Qualification tests using sine waves are covered by Specifications MIL-E-5272 and MIL-STD-810, Method 514. Fragility tests using sine waves are illustrated in figure 4-21, from which profiles of malfunction are developed for equipment, as shown earlier in figure 4-2.

(2) Random vibration tests were developed to more closely approximate service environments encountered by missiles and high performance jet aircraft. In most cases, random tests of equipment are made to spectrum shapes specific to a missile, equipment location, and launch profile. General spectrum shapes for qualification and fragility tests are given in figure 4-22. The basic specification for random tests is given in MIL-STD-810, Method 514.

4-7. Dynamic performance history of equipment.

a. General. In the event that test data are not available for an item of equipment, or a unique type of equipment is required, the design schedule may not allow sufficient time to obtain test data before the item is chosen and the fragility design fixed. It is then necessary to estimate the fragility of the chosen equipment. The fragility can often be estimated by analysis of the equipment design or by careful comparison with other tested equipment having similar construction features (e.g. BAS, 1979).

b. Equipment categories. There have been several attempts to classify equipment in various categories according to function and fragility. One example of such a classification is found in NHA (1963) and is displayed in table 4-3. Only the lowest vulnerability level and equipment frequency are presented for each item category in the table; the equipment used in a typical facility will generally fall into one of these categories. The values shown in the table will usually lead to conservative shock isolation design requirements.

c. Weapon system equipment.

(1) Weapon system equipment and subsystems are for the most part specifically designed, procured, and tested to meet the threat criteria for a specific weapon system or protective facility. Equipments that have some common usage would have to be evaluated for a new protective facility in terms of their past dynamic test and qualification requirements. Additional shock or transient testing may or may not be required for new applications, depending on what is learned from examination of classified reports from the controlling Department of Defense agencies.

(2) Test data will usually be found in various government and manufacturer's files. Occasionally, a test report for a particular item is released through a government publication. Useful information on shock damage from battle conditions may be seen in chaper 2 of NAVSHIPS (1964). It may be noted in this reference that for German "U" boats forced to surface from depth charges in World War II, the cause was electrical equipment damage 44 percent of the time.

(3) Current and planned protective facilities are a source of potentially useful equipment fragility infor-

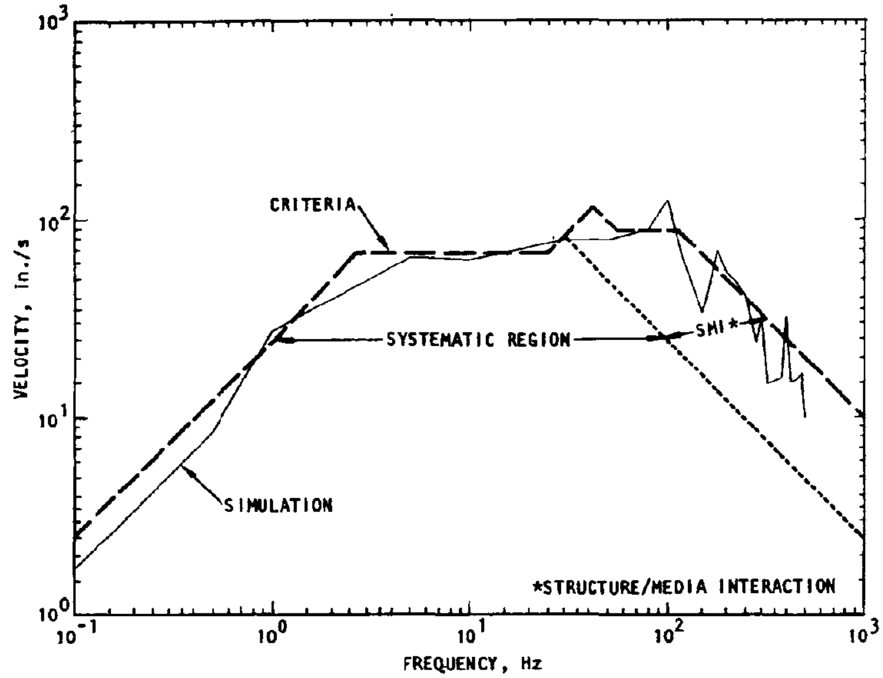


FIGURE 4-15. 46C COMMUNICATION REPEATER: VERTICAL BASE-LINE CASE SHOCK SPECTRA (Safford, 1974)

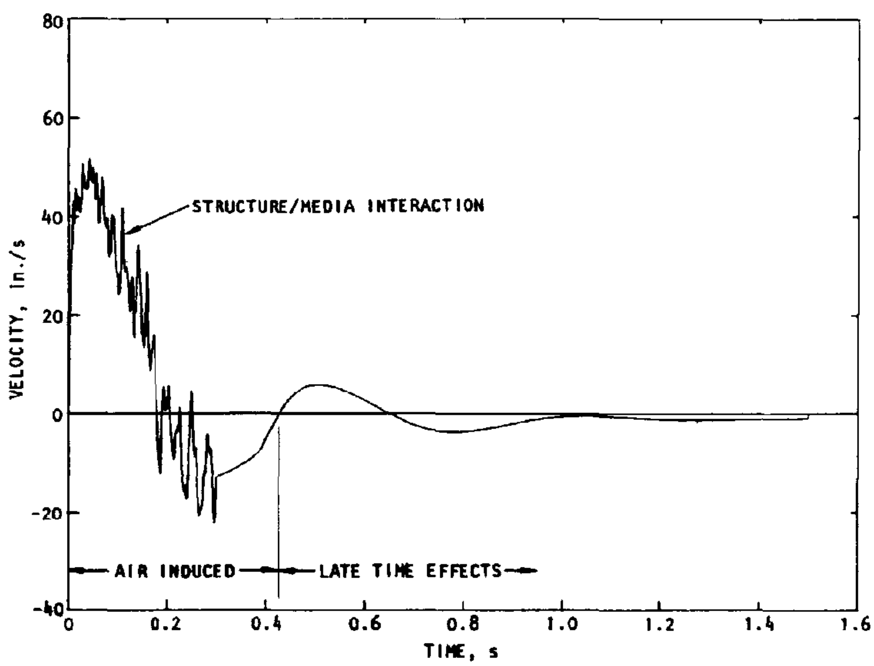


FIGURE 4-16. 46C COMMUNICATION REPEATER: BASE-LINE CASE VELOCITY (Safford, 1974)

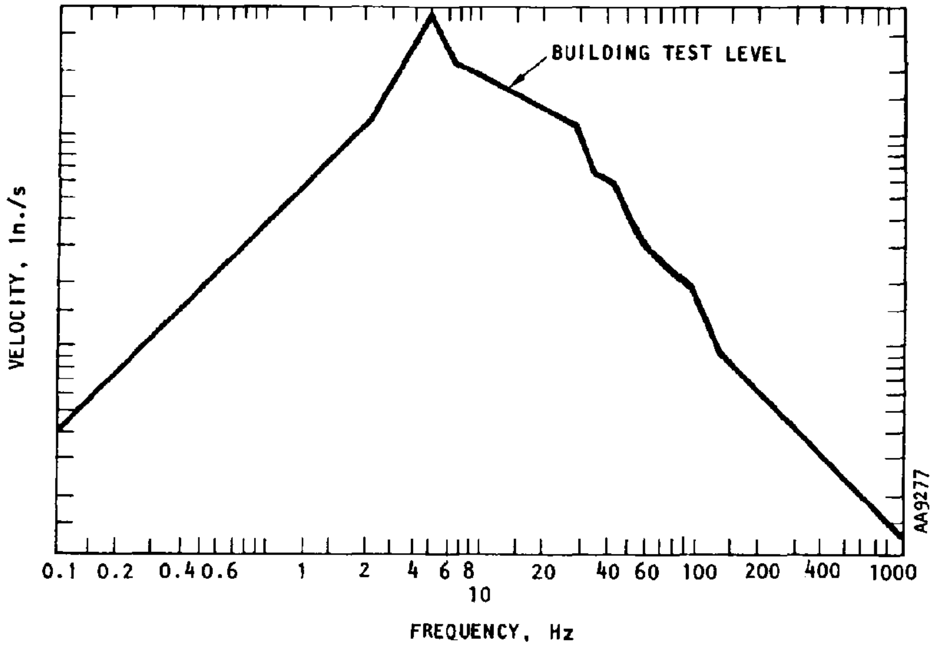


FIGURE 4-17. INPUT CRITERION FOR COMMUNICATIONS SYSTEM (Vertical, undamped) (Safford, 1974)

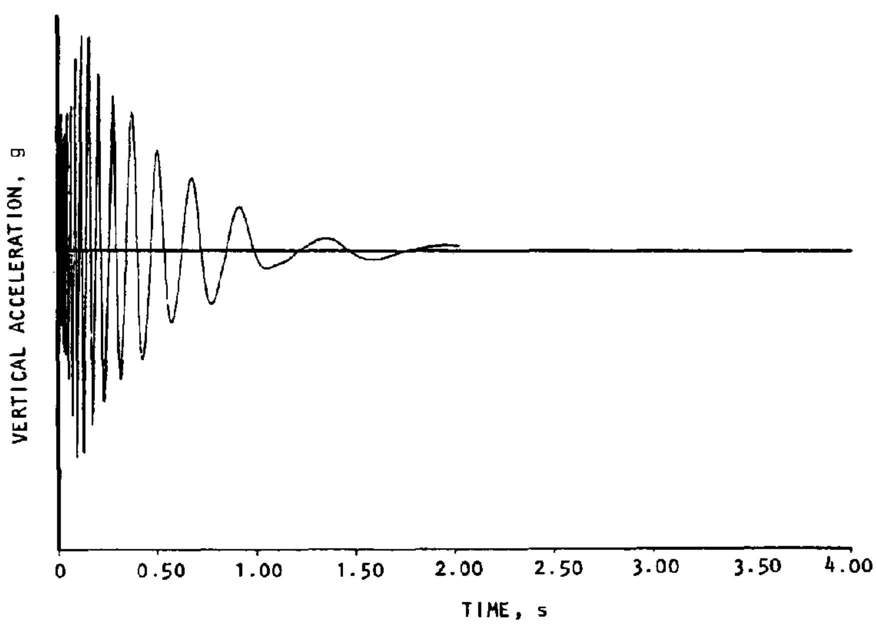


FIGURE 4-18. TEST VERTICAL ACCELERATION WAVEFORMS BY SINE BEAT WAVE SIMULATION TO MATCH SHOCK SPECTRUM CRITERION OF FIGURE 4-17 (Safford, 1974)

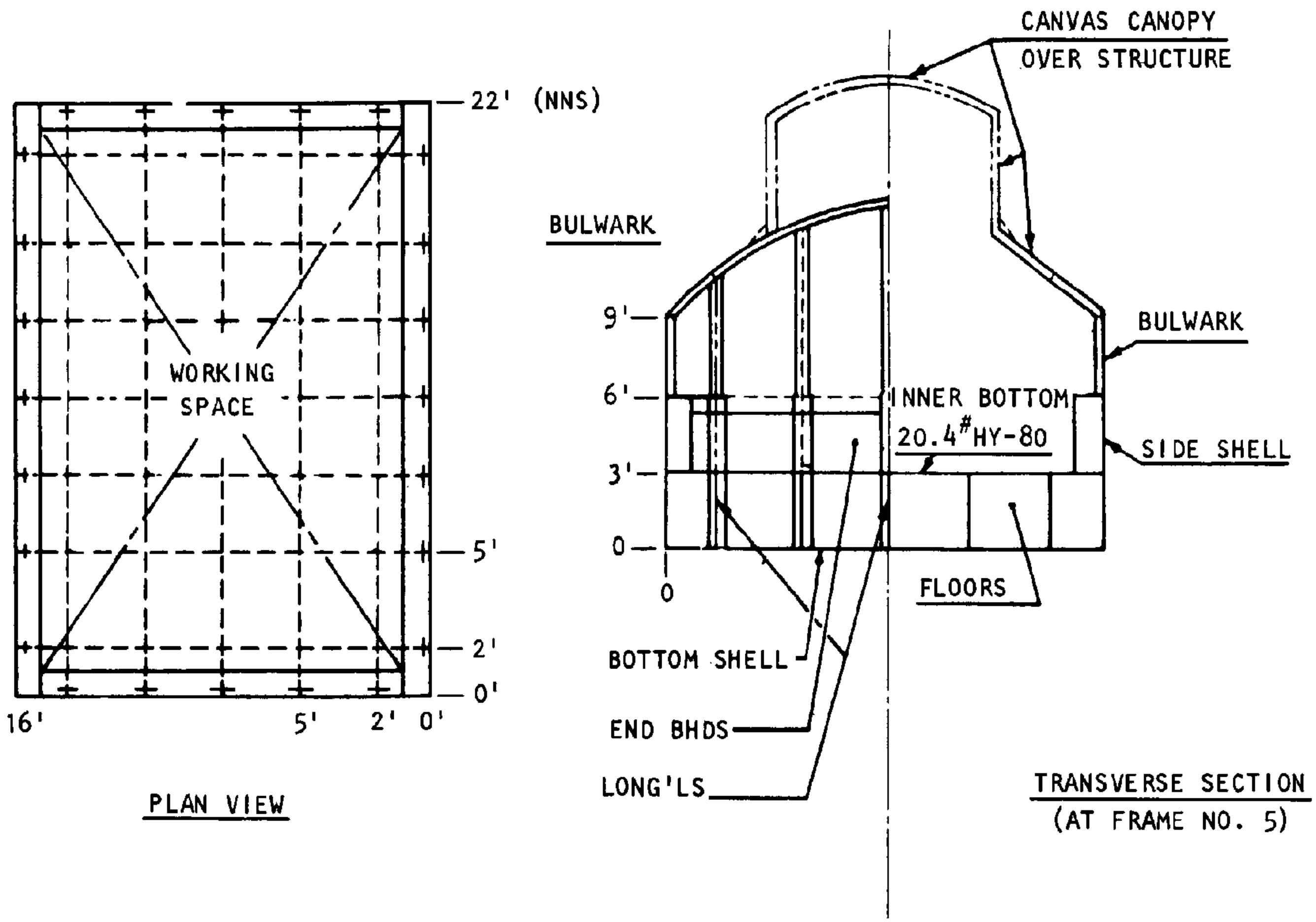


Figure 4-19. Floating Shock Test Platform (Navy, 1963)

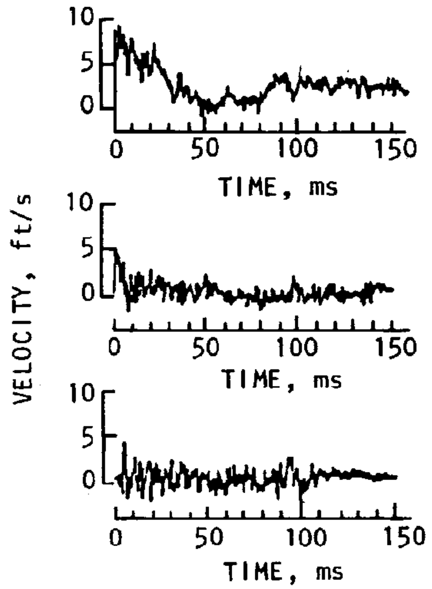


Figure 4-20. Typical Velocity Waveforms Measured on the Mounting Plane of the Large Floating Shock Platform (Clements, 1974)

mation. Such protective facilities include the following:

—Titan
—Minuteman and Minuteman upgrade
—Safeguard (phased out)
—North American Defense (NORAD)
—Alternate National Military Command Center (ANMCC)
—Sanguine
—MX system
—Naval vessels

These systems provide protection for equipment that is hard mounted to the facility, individually shock isolated, and shock isolated in equipment groups on platforms. A common threat characteristic of the systems is transient shock loadings transmitted by the facilities to the equipment and weapons. These threats vary for each facility and may change (increase) due to weapon improvements.

(4) Although very few summaries of equipment

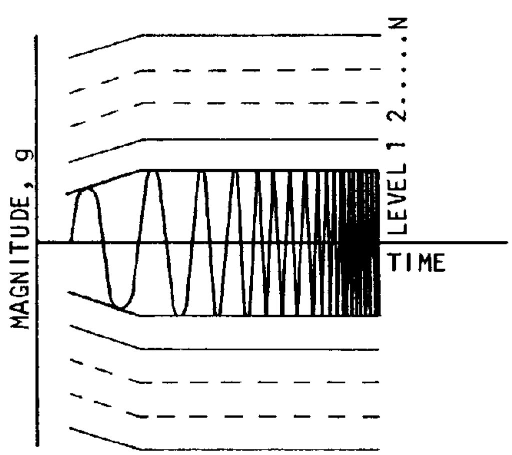


Figure 4-21. Slow Sine Sweep and Dwell Fragility Test (Rountree-Safford, 1979)

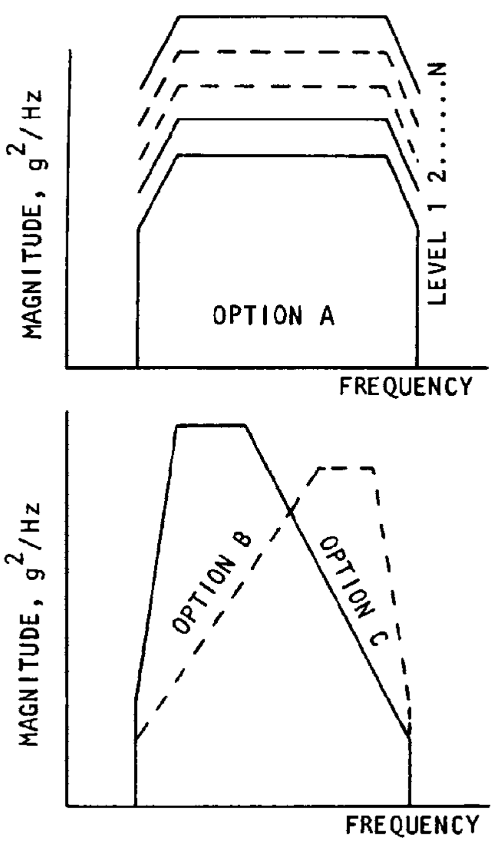


Figure 4-22. Spectrum Shapes for Random Vibration (DOD, 1959)

test data have been attempted because of the magnitude of the effort required and the rapid obsolescence of most equipment, summaries of test results obtained from a number of government agency reports are given in appendix A. It is possible to refer to the reports cited in the tables of the appendix for more accurate descriptions of the equipment involved.

d. Commercial equipment.

(1) Commercial equipment is used in varying amounts in protective structures (Safford, 1974; Dembo-Huang, 1969; and Boyd-Huang, 1977). Except for nonoperating transportation and handling (Dembo-Huang, 1969), commercial equipments are not designed to withstand dynamic transient shock loadings to be expected in weapon-system protective enclosures. Some classes of new equipments such as electrical power and control apparatus are being designed to meet earthquake seismic requirements (IEEE, 1975). Earthquake and nuclear weapon effects of ground shock and airblast are also being considered for new installations of telephone equipment (Rempel, 1967). Apparatus and equipment used aboard commercial deep-water ships may also be considered to have some degree of environmental hardness.

(2) The most serious problem in the application of commercial equipment is the susceptibility of electrical, mechanical, and optical components and subsystems to transient shocks. Analytical modeling of these devices is difficult and costly. However, in some cases, practical solutions have been achieved by environmental tests. Many component/part manufacturers pass out shock and vibration data on their products.

(3) A great deal of hardness upgrade on commercial equipment may be accomplished by inspection and from experience and judgment as to obvious deficiencies. A partial list of susceptible elements is given below:

- —Relays
- —Circuit breakers
- —Equipment drawers (no blocks)
- —Plug-ing modules (no blocks)
- —Tape drive units
- —Memory units and devices
- —Computer peripheral equipment
- —Cabling and wiring harnesses (poor and insufficient tie downs)
- —Hook-up wires (inadequate vibration loops)
- —Loose debris in equipment (electrical shock caused by washers, screws, metal chips, wire trimmings)
- —Large circuit board connectors (unsupported)
- —Large printed circuit boards (unsupported)
- —Component mounting screws and bolts (not properly torqued)

A number of references supplement the above listings with more detail and discuss numerous aspects of hardening equipment: Fischer, (1976, 1977), NAVSHIPS (1949, 1957), Navy, MIL-S-901, USAF (1957), DOD MIL-STD-810, Anonymous (1977), ANSI (1964), and Hua (1977).

(4) Hydraulic surge by fluids such as fuel, water, and temperature control systems through hydraulic lines to equipment must also be considered for potential damaging effects. Hydraulic surge applies to both hard-mounted and shock-isolated equipment. A description of hydraulic surge tests to fragility levels is given in Eckblad-Hedrick (1970).

4-8. Personnel tolerances.

a. General. Estimates of the reaction of the human body to structural motions resulting from a nuclear detonation are based on a variety of tests and on accident summaries. This broad range of information indicates that the effects on man depend on the magnitude, duration, rise time, frequency, and direction of motion, as well as body position and restraints. However, both experimental and analytical data on which to base human shock tolerances for existing underground protective structures were and still remain extremely limited. Data have been collected during tests ranging from long-lasting (1 to 3 sec duration) acceleration to very brief impacts resulting from falls or drops. Some of the pertinent data is summarized in the following paragraphs. In a few cases the data establish injury thresholds, but more often they relate to voluntary acceptance of pain.

b. Primary loadings.

(1) Most protective structures are designed to withstand ground shocks of sufficient strength to require shock protection for all personnel. Thus, the personnel are exposed most frequently not to the complex motion of the ground, but to the nearly sinusoidal damped oscillation of the isolated platform or other supporting device. Further, the basic design criteria for many facilities specify that the personnel will be assumed to receive no warning of attack and therefore may be occupied by any of their normal functions at the instant the shock occurs.

(2) Initial structural motions may be either up or down, and are normally accompanied by a simultaneous horizontal motion. In table 4-4 the maximum accelerations for personnel areas are shown for several weapon system facilities.

(3) Although the natural frequencies of the personnel shock isolation systems are not indicated in the table, in all cases they are less than one cycle per second. Similarly, the mode of failure in all cases is based on impairment of operational capability of all personnel, some of whom may be standing unsupported and may be unprepared. The downward trend in peak accelerations believed to be necessary to achieve the desired protection is clearly indicated by the differences

TABLE 4-3. ESTIMATES OF FREQUENCY AND VULNERABILITY OF TYPICAL EQUIPMENT ITEMS (NHA, 1963)

Item	Lowest Fundamental Natural Frequency, Hz	Estimated Vulnerability Level, g
Heavy machinery--motors, generators, transformers, etc. (>4000 lb)	5	10
Medium weight machinery--pumps, condensers, air conditioning, etc. (1000 to 4000 lb)	10	15
Light machinery--fans, small motors, etc. (<1000 lb)	15	30
Racks of communication equipment, relays, rotating magnetic drum units, large electronic equipment with vacuum tubes	10	2
Small electronic equipment--radios, incandescent lamps, and light bulbs	20	20
Cathode ray display tubes	5	1.5
Transistorized computers, fluorescent lamps and fixtures, nuclear reactors	10	5
Storage batteries (all types), piping, and duct work	5	20

between those specified by early design criteria and thos used in final design of the Titan II and Minuteman facilities.

(4) It should be noted that all the suspension systems indicated are pendular and that the maximum horizontal acceleration is fixed more by the practical aspects of the pendulum design than by a human shock tolerance.

c. *Human tolerance to impact.*

(1) Sensitivity diagrams

(a) Figures 4-23 and 4-24 summarize observational data in terms of the coordinates of a sensitivity diagram: average accelerations a as abscissas, and speed change or impact speed v as ordinates. Since scales of both coordinates are logarithmic, loci of constant duration are straight lines of slope + 1.

(b) Points corresponding to the conditions of observed or inferred impacts are collected in figure 4-23, and conclusions drawn from these data are represented in figure 4-24 by "safe" and "unsafe" regions. When *both* speed change (ordinate) and average acceleration (abscissa) are higher than certain limits, the

TABLE 4-4. MAXIMUM ACCELERATION IN WEAPON SYSTEM PERSONNEL AREAS (Parsons, 1962)

Weapon System	Maximum Acceleration in Personnel Areas, g	
	Vertical	Horizontal
Atlas Silo	1.5	0.125
Atlas Control Center	"... mounted to reduce ground shock without impairing operational ability"	
Titan II (Criteria)	3.0	3.0
(Initial Design)	2.4	0.5
(Revision)	0.5	0.5
Minuteman (Criteria)	1.0 (down) 3.0 (up)	1.0
(Design)	0.5	0.15

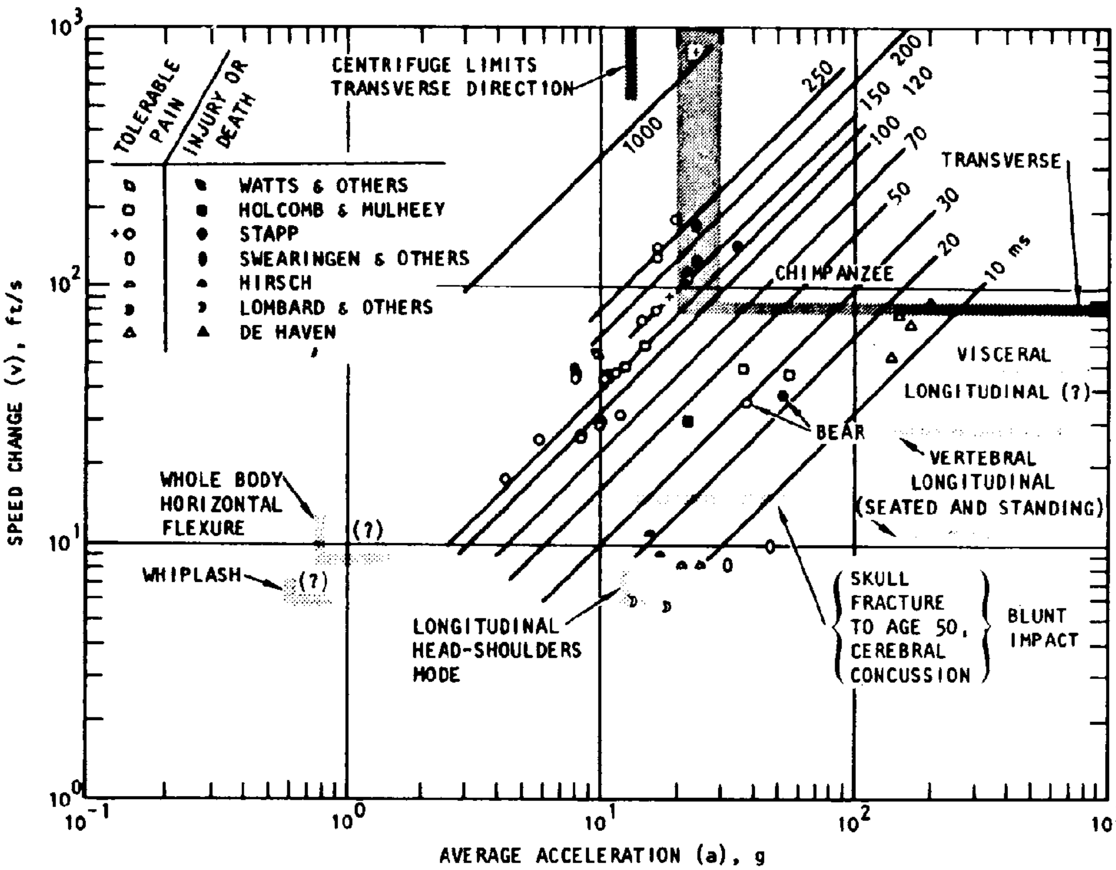


Figure 4-23. Human Tolerance Limits to Impact (Rempel, 1967)

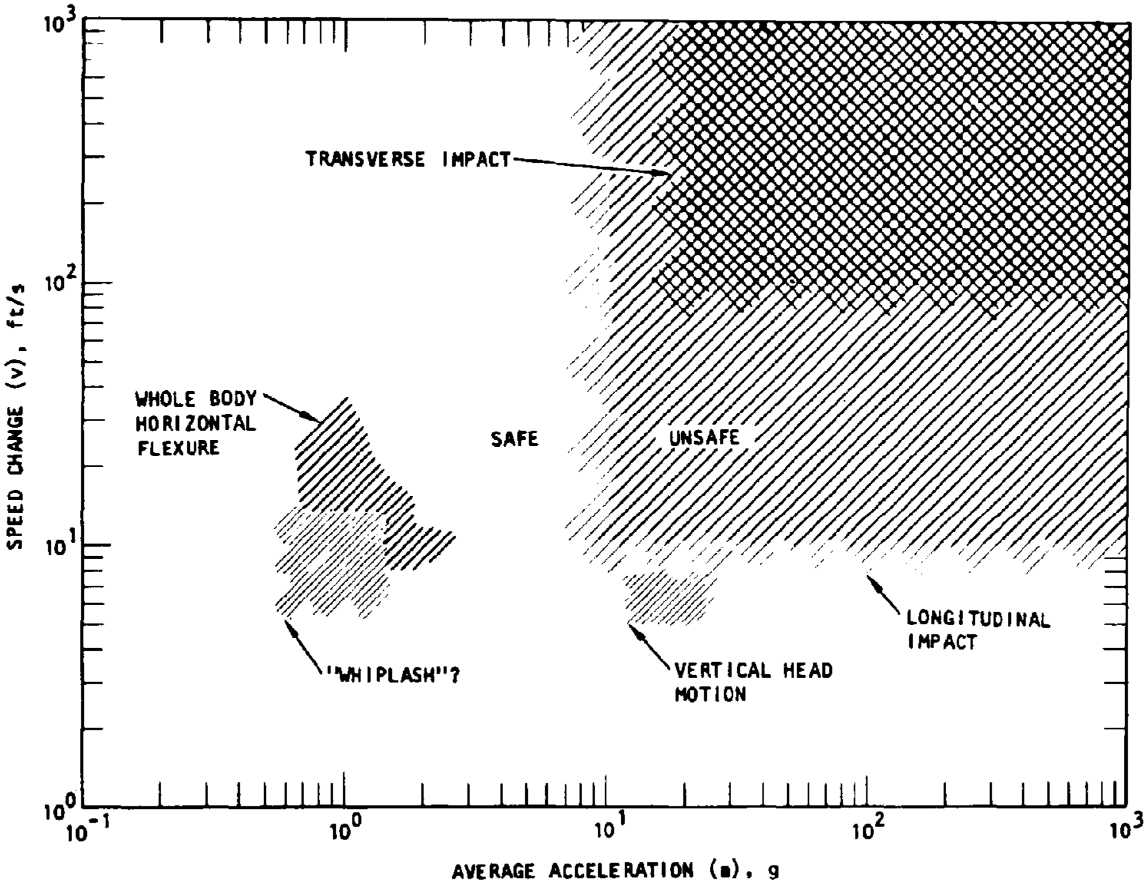


Figure 4-24. Human Tolerance to Impact (Rempel, 1967)

impact is unsafe; otherwise it is safe. For both theoretical and observational reasons, these limits, particularly the average acceleration, cannot be given as single values and are sometimes shown in the figures as regions.

(c) The limits depend on bodily orientation with respect to the acceleration vector and the various bounds drawn in the figures labeled to indicate the mode of impact. "Longitudinal" means acceleration is applied to the whole body parallel to the spine; "transverse" indicates the vector is normal to the spine. "Whiplash" is a motion of the head with respect to the shoulders either fore and aft or sidewise; the limits shown in the figures for whiplash are not well established.

(d) Drop tests were conducted on a standing man (Gesswein-Corrao, 1970), which generated force-time history curves as shown in figures 4-25 and 4-26. These drop tests (velocity shock) consisted of dropping a standing man on a small platform from a known height to impact on a force gage.

(e) Figure 4-25 shows a curve that illustrates the general results obtained in all tests. There occurs first a compression phase in which drop energy results in a force increase to a maximum. The time to peak force decreased from about 13 msec for a 2 in. drop to about 7 msec for a 9-in. drop. Then the body recovers from the impact towards its normal weight. This time varied between 30 and 60 msec. It seems significant that in all the tests conducted only twice did the rebound force ever reach zero. Apparently, the body defends against impact largely by energy dissipation rather than by energy storage.

(f) The test result judged most significant to tolerance studies is shown in figure 4-27. Here the two highest values of peak force developed above body weight are plotted against the kinetic energy of each volunteer at impact. The straight line represents the least squares fit to the data that had the highest index of determination (0.85) of several curves types tried. The equation is the fitting equation

$$F = 4E^{0.85} \qquad (4\text{-}8)$$

where E is impact energy in in.-lb and F is peak force in pounds above body weight.

(g) The time interval between foot force and head acceleration against the kinetic energy of impact is plotted in figure 4-28. The transit time does not seem to be greatly influenced by height of drop. Apparently, stress transmission occurs at a rather constant velocity, somewhat less than 1000 ft/sec. This

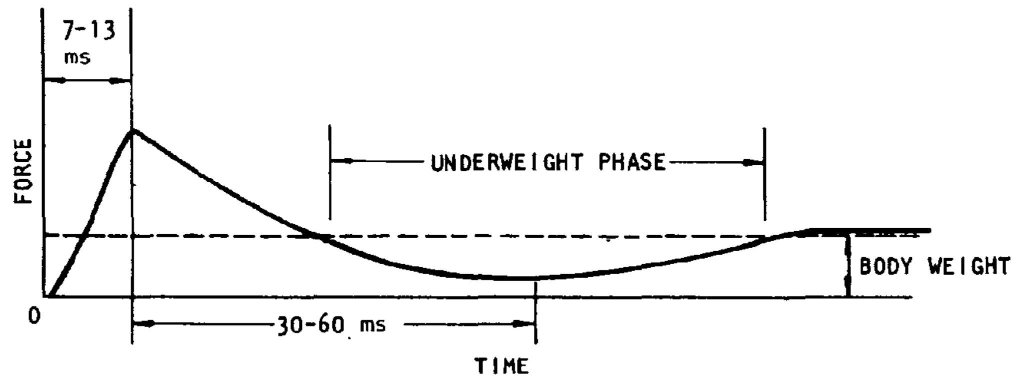


FIGURE 4-25. TYPICAL FEATURES OF FORCE-TIME CURVES (Gesswein-Corrao, 1970)

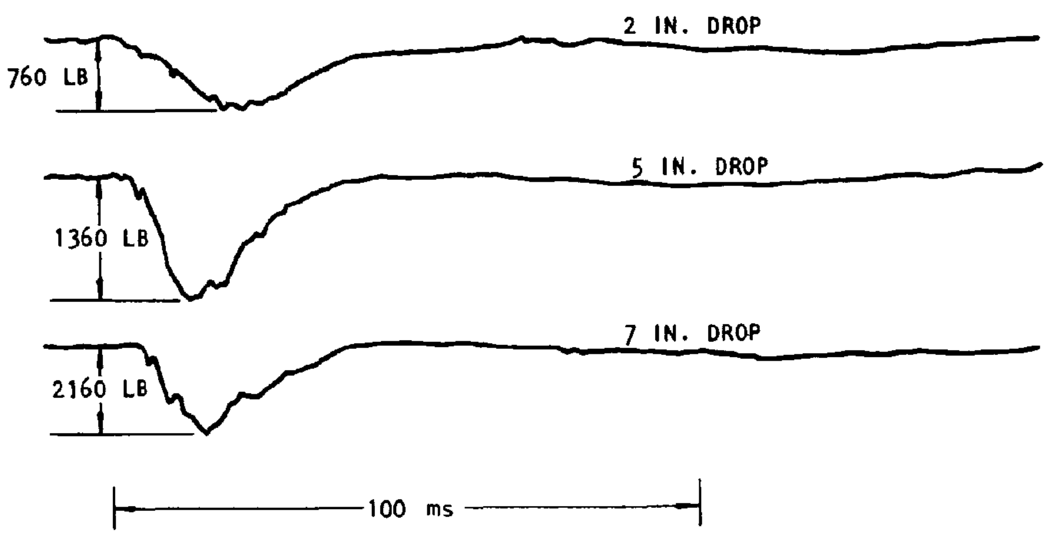


FIGURE 4-26. FORCE-TIME CURVES FOR ONE VOLUNTEER: 2,5, AND 7-INCH DROPS (Gesswein-Corrao, 1970)

speed is rather low, considering that bone transmits sound at a velocity near 10,000 ft/sec and the soft body materials (mostly composed of water) at a speed near 5000 ft/sec. It would seem that the body's response to impact cannot be readily explained by assuming longitudinal stress wave propagation as in an elastic medium. Apparently, joint distortion with its accompanying generation of transverse shear waves slows elastic propagation times considerably.

d. Effect of body position

(1) The variation in man's shock tolerance for different body positions, as a function of duration of shock acceleration, is shown in figure 4-29. Curve D in the figure is an upper design limit, above which injury can occur. Curve E is based on subjective comfort response to centrifuge tests.

(2) Because of the possibility of impact injuries to personnel, design criteria will often limit acceleration inputs to much lower values than those indicated in figure 4-29. Goldman and Von Gierke (1960) suggest

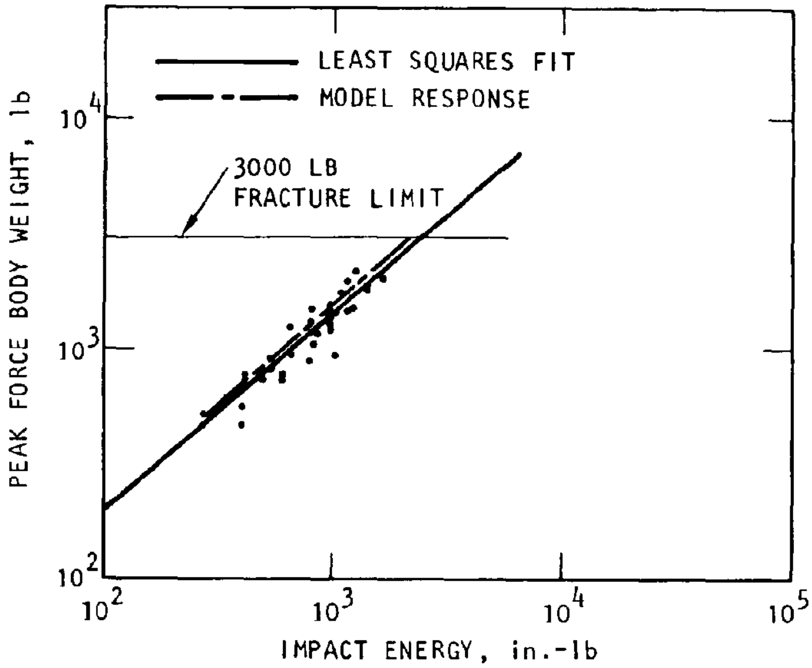


FIGURE 4-27. PEAK FORCE DEVELOPED ABOVE BODY WEIGHT VERSUS IMPACT ENERGY (Gesswein-Corrao, 1970)

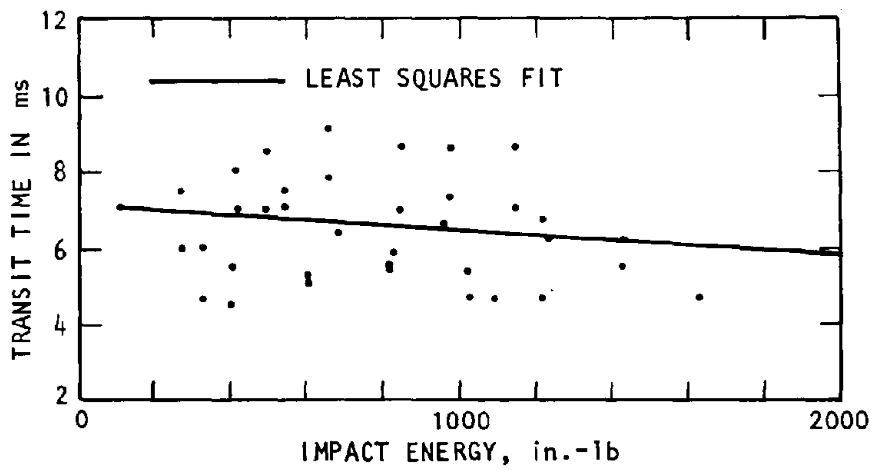
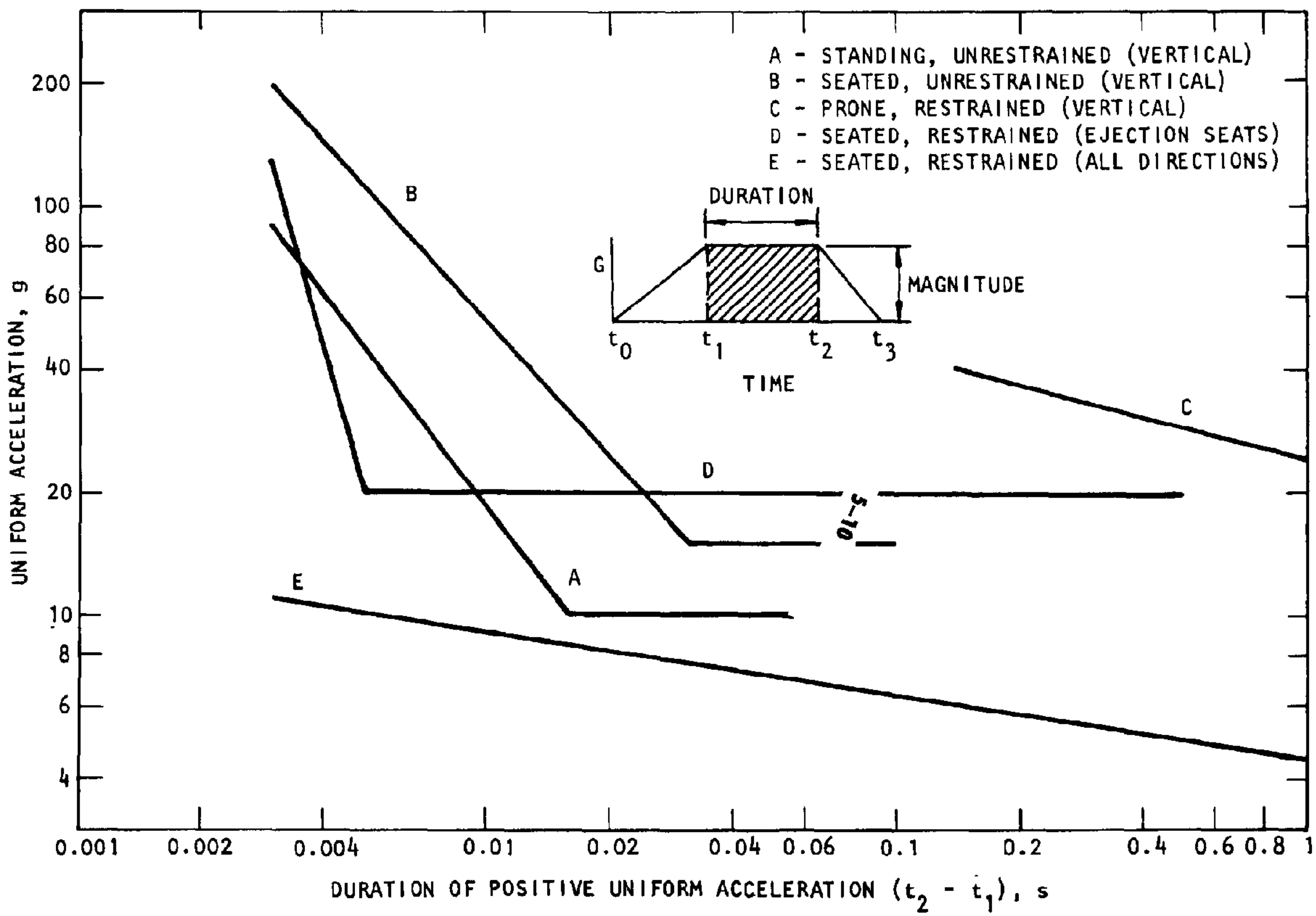


FIGURE 4-28. STRESS WAVE TRANSIT TIME VERSUS IMPACT ENERGY (Gesswein-Corrao, 1970)



NOTE: REFER TO PARAGRAPH 4-8d FOR INTERPRETATION AND APPLICATION

Figure 4-29. Human Shock Tolerance to Prevent Internal Injury (OCE, 1966) limiting upward floor accelerations to 0.75 g and downward accelerations to 0.5 g to avoid injuries to an unrestrained man in the standing position and 0.75 g upward and 1.0 g downward for an unrestrained man in the sitting position. Horizontal acceleration limits of 1.0 g are also recommended for an unrestrained man in the sitting position to avoid whiplash injuries or his being thrown from his chair. Curves A and B of figure 4-29 are applicable to the standing and sitting unrestrained positions, respectively, and represent upper design limits. These curves are appropriate for design only under conditions where impact injuries can be avoided. Values from curve A represent one-half of the shock tolerance limit to allow for the possibility that all of the subject's weight might be on one leg.

(3) An unrestrained man in the prone position (applicable to sleeping quarters) would be expected to withstand higher accelerations, except that he may be

TABLE 4-5. RECOMMENDED DESIGN VALUES FOR HUMAN TOLERANCE TO SHOCK MOTIONS (IN G'S) (Coldman-Von Gierke, 1960)

Criterion A = Limiting g value to prevent internal injury

B = Limiting g value to avoid possible impact injury

	Posture	Criterion	Shock Direction: Upward	Downward	Horizontal
Unrestrained	Standing	A	10.0	5 in. at 1 g	*
		B	0.75	0.5	0.5
	Sitting	A	15.0	19 in. at 1 g	*
		B	0.75	1.0	1.0
	Prone	A	40.0	*	*
		B	0.75	1.0	0.75
		-	Curve C (Fig. 4-29)	Curve C	Curve C
Restrained	Sitting	A	15.0 or Curve E (Fig. 4-29)	15.0 or Curve E	15.0 or Curve E
	Prone	A	40.0 or Curve C (Fig. 4-29)	40.0 or Curve C	15.0 or Curve E

*Displacement must be limited, i.e., attenuated to less than the distance between operator and nearest solid object.

thrown from the bed and sustain injuries upon impact with the floor. For this reason, design criteria are again lower than human shock tolerance; acceleration limits of 0.75 g upward, 1.0 g downward, and 0.75 g horizontal are suggested. If the man is in the unrestrained prone position, but inside a completely padded enclosure, he would be partially protected from impact injuries and curve C of figure 4-29 is applicable. If the subject is in a sitting position and restrained to prevent his leaving the seat, and if relative motion between head and limbs and body is prevented, curve E of figure 4-29 is applicable for upward, downward, and horizontal motions. Curve D is used for design of upward ejection seats and does not include the effect of rapid deceleration. Curve C is applicable to upward or downward acceleration for the prone restrained position and curve E for horizontal motions.

(4) Based on figure 4-29, table 4-5 summarizes the recommended human tolerance limits of shock acceleration for danger in protective structures. The lower of the recommended limits for an unrestrained man are conservative to avoid possible impact injury caused by loss of balance. Restraints for seated and prone positions are an obvious advantage and may be required where practical and consistent with mission functions.

(5) It is difficult to isolate the effect of the rate of rise to the peak shock acceleration on man. However, Eiband (1959) indicates that a rate of rise equal to 1000 g per sec can be tolerated at the shock levels recommended for design. In designs where vertical and horizontal shock pulse effects are combined, it is recommended that the acceleration vector sum be limited to the lower of the horizontal or vertical limits.

(6) The tolerance curves in figure 4-29 may be interpreted and adjusted by considering the probability

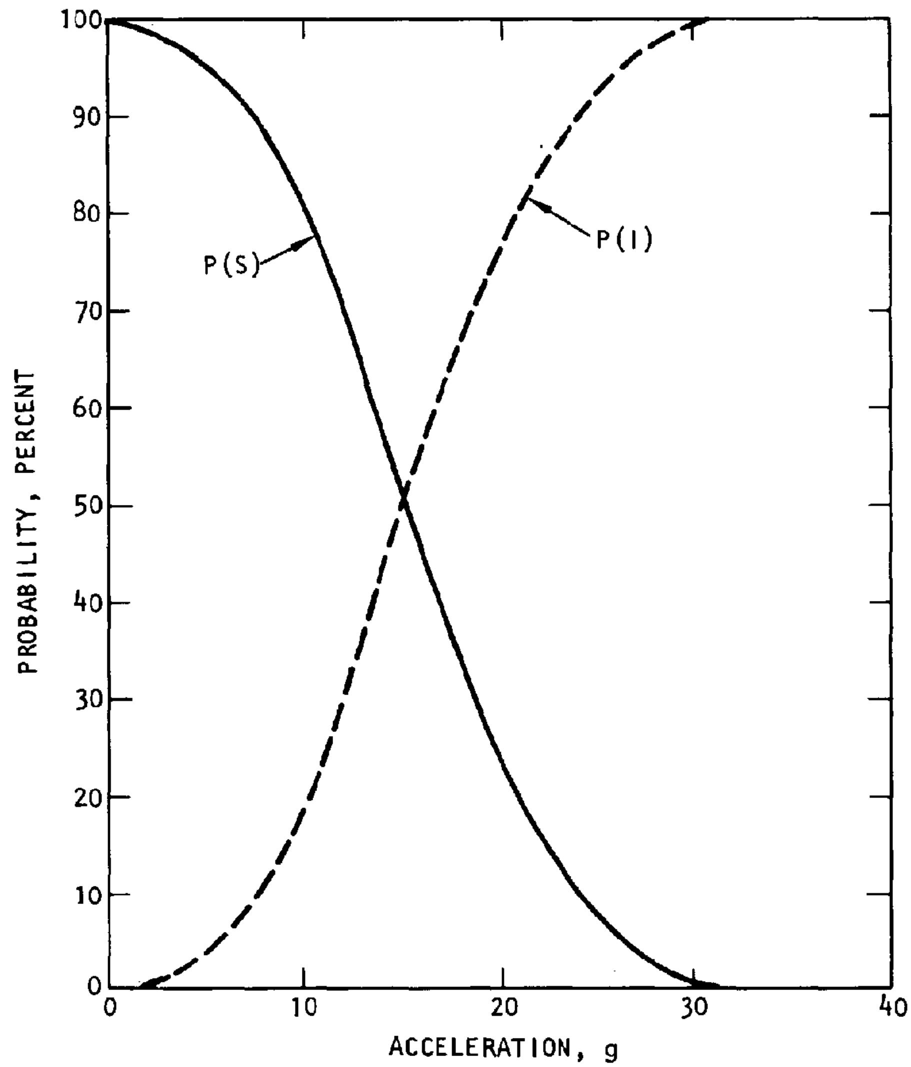


Figure 4-30. Probability of Survival (S) and Injury (I) vs. Upward Vertical Acceleration Standing, Unrestrained (BAC, 1977)

of survival (or injury) versus upward vertical acceleration for an unrestrained man. These probabilities are given in figure 4-30 for a standing man and in figure 4-31 for a seated man.

(7) Additional tolerance information assembled from several sources is given in figure 4-32 and 4-33. These figures are in the form of acceleration/pulse-duration plots for a seated person. They show the parts of the body that are the limiting factors for different pulse durations.

(8) Conclusions reached in the study of Hirsch (1964) for an unrestrained standing and seated man are:

—A man standing with weight equally distributed on both legs is not likely to be injured when the change in peak velocity is less than 11 ft/sec and the rise time to peak velocity is between 1 and 20 msec.

—A seated man is not likely to be injured when the change in peak velocity is less than 15 ft/sec and rise time to peak velocity is between 1 and 20 msec.

—A standing man is more vulnerable than a seated man in the 1- to 20-msec time duration regions of shock motion.

—Man will leave the deck and fly upward with kickoff velocities between 60 and 120 percent of the peak deck velocity.

(9) Where restraints in a seated and prone body configuration are required, both chair and bed supports should include mechanical fuses that transmit essentially constant force to the occupants. These devices may be frangible tubes or the somewhat more sophisticated rolling ring absorber (Platus, 1973). Constant force liquid absorbers equipped with spring return may also be obtained (from Taylor Devices Inc., North Tonawanda, N.Y., for example.)

e. Injuries

(1) Data indicate that man can tolerate relatively large acceleration forces when he is restrained. However, the unrestrained person is subject to loss of balance at relatively low accelerations with the attendant possibility of impact injury.

(2) Man may strike overhead objects, lose footing, and fall against objects or the floor. Goldman and Von Gierke (1960) indicate that skull fractures occur if the head hits a hard flat surface after a free fall of 5 ft. Head impact with a blunt corner requires only a small fraction of the 5 ft drop for a skull fracture. If the chance of impact injury for the unrestrained operator

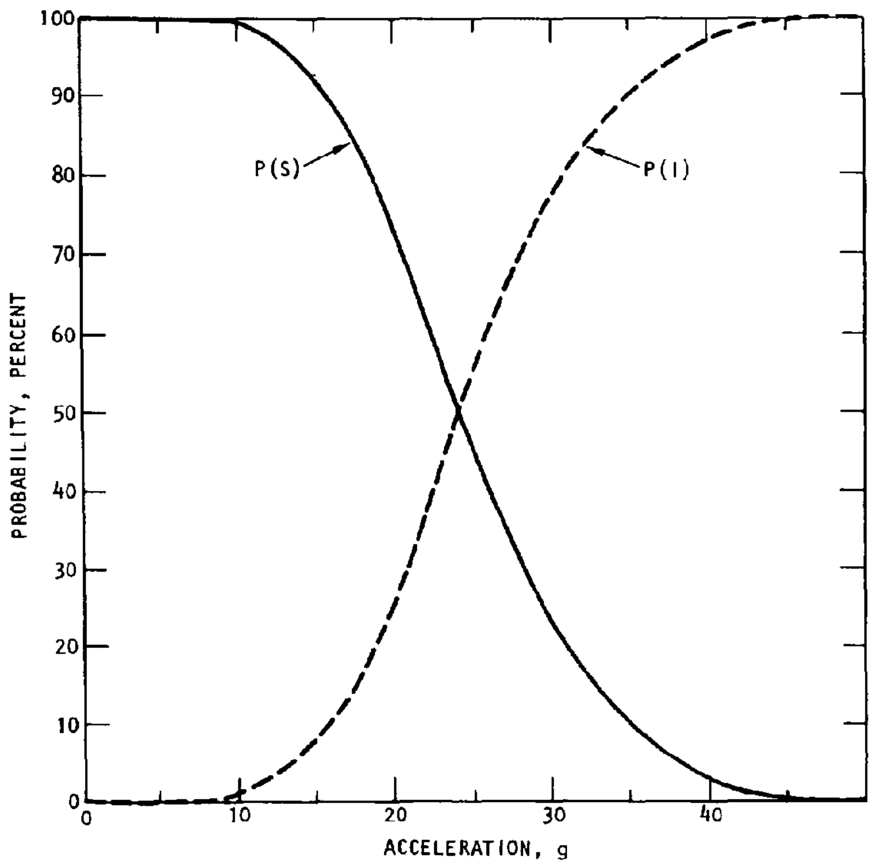


Figure 4-31. Probability of Survival (S) and Injury (I) vs. Upward Vertical Acceleration Seated, Unrestrained (BAC, 1977)

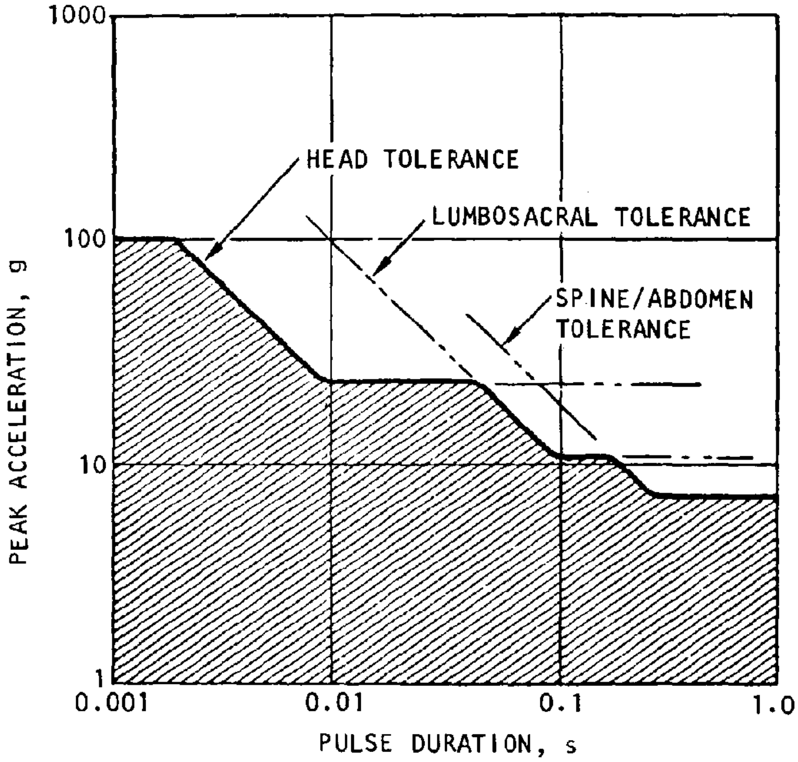


Figure 4-32. Composite Tolerance for Upward g Acceleration While Seated (BAC, 1977)

can be eliminated by the use of protective gear, such as crash helmets, and by providing resilient padding on all exposed corners and projections, the accelerations indicated by curves A and B in figure 4-29 can be used for design maxima.

(3) Injuries can result from excessive acceleration of the body. Hirsch (1964) concludes that a standing man will receive compressive injuries in the body-supporting bones if the upward floor acceleration exceeds 20 g during a long-duration loading. The injury threshold is higher for short-duration loads. If the floor moves downward rapidly, the man will free-fall a maximum distance equal to the floor displacement. A drop of only 20 in. (50.8 cm) could fracture both legs. Impact with an upward moving floor would increase the probability of fracture. Damage to the vertebral column is a common mechanism of injury where impact is parallel to the spine (Ruff, 1950). To mitigate injuries due to deck accelerations, the U.S. Navy developed shock attenuating deck pads. These are described in reports by Hawkins and Hirsch (1966 and 1968).

(4) Studies of seated and standing unrestrained humans at David Taylor Model Basin (Navy Ships Research and Development Center) covered effects for simulated vertical shock motion of ship decks (fig. 4-34). Generally, with a constant input upward vertical velocity amplitude, the kickoff velocity (unrestrained man) increases as the time to reach peak velocity increases. A means of predicting man's kickoff velocity for the standing stiff-legged and seated positions (Mahone, 1966) can be developed as

$$V_K/V_O = 1.5\,(t_p/T)^{0.256} \qquad (4\text{-}9)$$

where

V_K = Subject's kickoff velocity, ft/sec
V_O = Peak input velocity, ft/sec
t_p = Rise time to peak input velocity, msec
T = Natural period of man, msec (T equals 100 and 167 msec for the standing and seated man, respectively; see Hirsch, 1964)

(5) Limited data are available from accident studies on injury severity. Accident reports of the American Alpine Club were evaluated by Stech and Payne (1969), using the severity scale developed by Cornell Aeronautical Laboratories. These data included few accidents involving head impact and, in general, these accidents resulted in a higher injury severity rating than equivalent accidents where no head impact was involved. The results of this study are shown in figure 4-35, where the midpoint of the injury scale separates 50 percent probability of injury or a velocity shock of 53 ft/sec. The data for the 50 percent probability of injury at 53 ft/sec impact is plotted in the form of acceleration/pulse-duration in figure 4-36.

(6) The free-fall accident data for mountain climbers is conditioned by minimum head injuries, impacts along the body length, impact anticipation, and bent

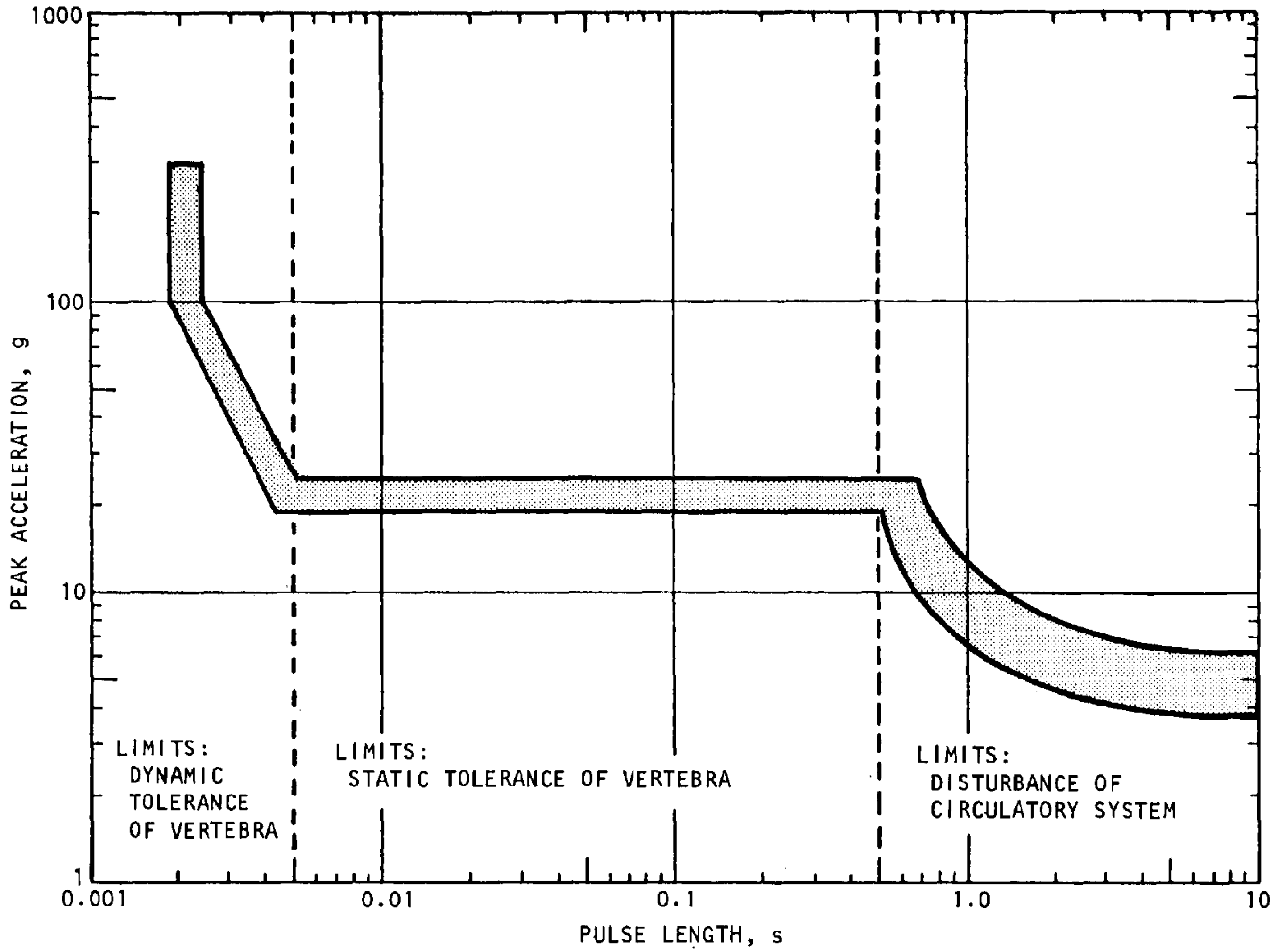


Figure 4-33. Effect of Seated Acceleration in the Direction Seat-to-Head (BAC, 1977)

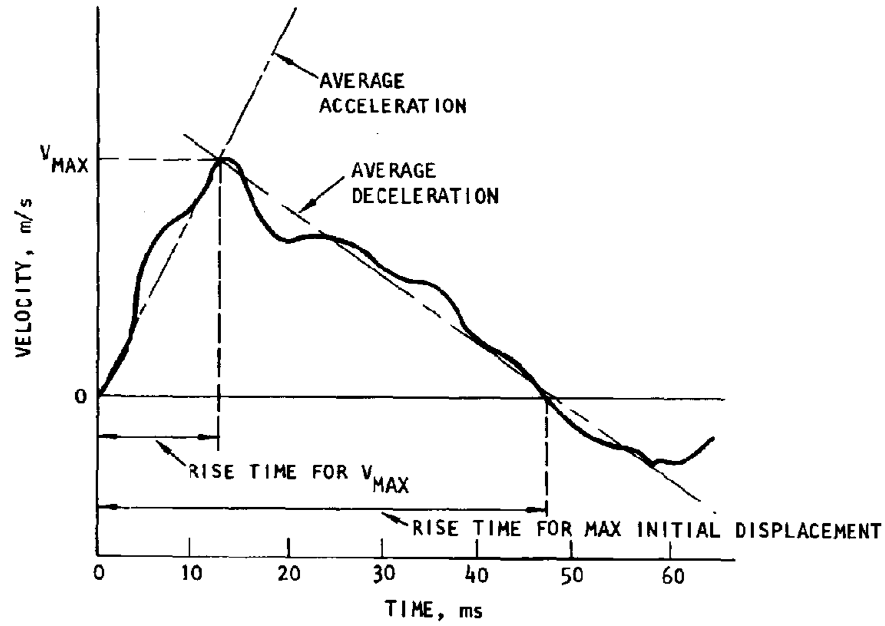


Figure 4-34. Example of Shock-Motion Profile and Terminology (Hirsch, 1964)

knees. These data are very optimistic in comparison to the data plot of percent injuries (combat ineffectiveness) as a function of pulse deck (ship) velocity given on pages 18 and 19 of the GE TEMPO report (1972). The injury curve of this reference should be used in conjunction with those given in figure 4-29.

(7) Higher shock loadings into the incapacity range have been studied in animals such as chimpanzees, Himalayan bears (126 lb), and mice. Mice have proven useful in obtaining ranges (5 to 95 percent) of incapacitating injuries for the same test load and relating injury to rise time of the shock load (Stech-Payne, 1969).

(8) Mitigation of personnel injuries may be increased by the addition of pads on the shock isolated platform made of lead-asbestos or urethane foam (Hawkins, 1968; Hawkins and Hirsch, 1966).

f. Vibration

(1) Where human performance is mandatory during or immediately after an attack for the continued functioning of a weapons system, the effect of vibrations must be considered.

(2) Human shock tolerance to steady-state vibrations may be of interest in those instances where there is little damping of shock-isolated platforms. As in the case of transient shock, tolerance of steady-state vibrations is affected by position and restraint of the individual. Vibration frequencies in the region of 5 to 10 Hz are of greater significance, since they approximate natural frequencies of the human body. At these frequencies, tolerance of steady-state vibrations is shown in table 4-6.

(3) Human performance can be degraded due to pain, respiration difficulties, disorientation, and lack of visual acuity. Studies on this subject are covered in Harris (1965); Edwards-Lange (1964); Weis (1966); Mandel-Lowry (1962); Ashley (1970); and DOD (1974). Figure 4-37 from DOD (1974) provides a suitable guide for shock isolation. Shock-isolated platforms should be timed to about 1 Hz rigid platform motion, since this frequency corresponds to the general human walking gait.

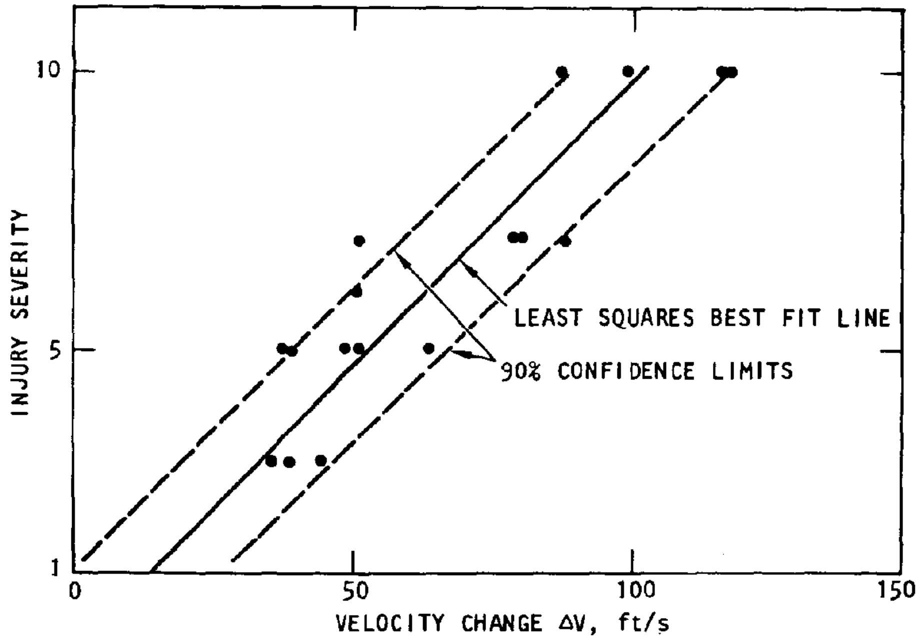


FIGURE 4-35. CALCULATED BEST FIT LINE AND 90 PERCENT CONFIDENCE LIMITS FOR INJURY OF SEVERITY VERSUS ΔV (Stech-Payne, 1969)

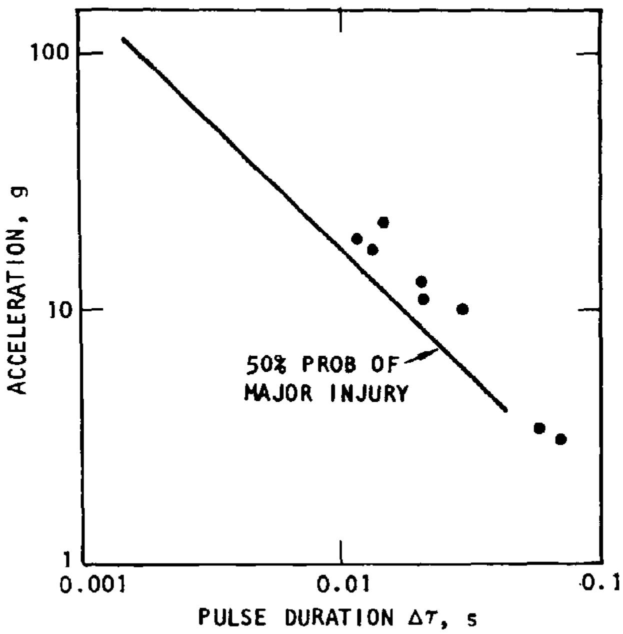


FIGURE 4-36. MAJOR INJURY LEVEL AT 50 PERCENT PROBABILITY FOR 53 FT/S IMPACT VELOCITY CONVERTED TO ACCELERATION-PULSE DURATION (Stech-Payne, 1969)

TABLE 4-6. RECOMMENDED DESIGN VALUES FOR HUMAN TOLERANCE TO VIBRATION (OCE, 1966)

Criterion A. = Limiting g value to prevent internal injury

Criterion B. = Limiting g to avoid impact injury

Posture		Criterion	Vibration Direction: Vertical	Vibration Direction: Horizontal
Unrestrained	Standing--Unrestrained	B.	0.5 g	B. 0.5 g
Unrestrained	Sitting--Unrestrained	B.	0.75 g	B. 1.0 g
Unrestrained	Prone--Unrestrained	B.	0.75 g	B. 0.75 g
Restrained	Sitting--Restrained	A.	Figure 4-29 3 g, f < 2 Hz 1 minute or less	A. Figure 4-29 3 g, f < 1 Hz 1 minute or less
Restrained	Prone--Restrained	A.	Figure 4-29 3 g, f < 2 Hz 1 minute or less	A. Figure 4-29 3 g, f < 2 Hz 1 minute or less

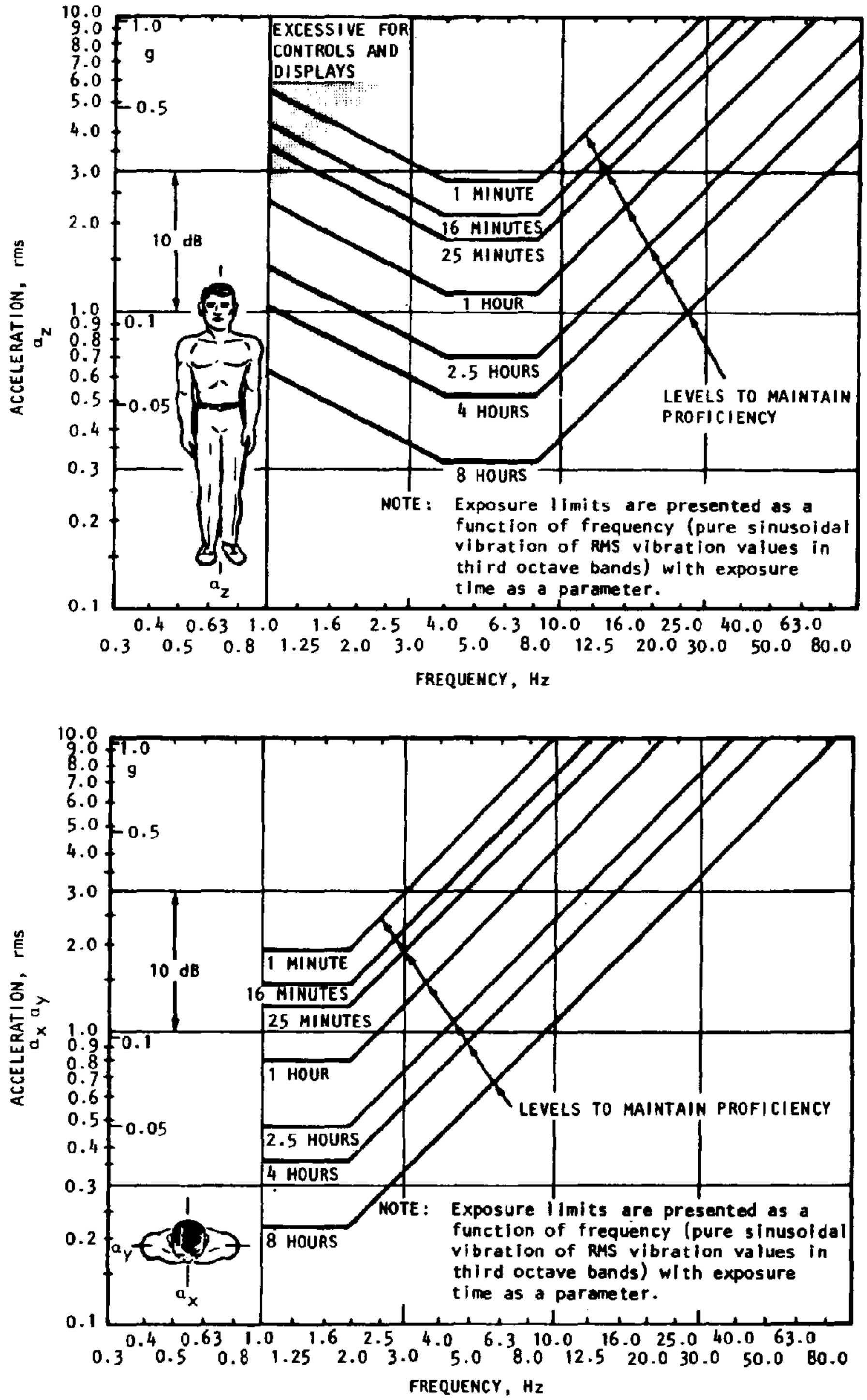


Figure 4-37. Vibration Exposure Criteria for Longitudinal (Upper Curve) and Transverse (Lower Curve) Directions with Respect to Body Axis (DOD, 1974)

CHAPTER 5

PERFORMANCE CHARACTERISTICS OF SHOCK ISOLATION SYSTEMS

5-1. Introduction.

a. Previous chapters have been concerned with facility environments and the ability of equipment and personnel to withstand these environments. The present chapter describes the performance of shock isolation systems needed to protect equipment/personnel from anticipated facility environments. A necessary preliminary to this description is a discussion of whether and to what degree shock isolation is actually required.

b. The description of shock isolation system performance begins with a discussion of single-degree-of-freedom systems. Such systems are often useful for idealizing real systems, which may consist of complex combinations of numerous smaller components. The analysis of single-degree-of-freedom systems clarifies important shock isolation concepts such as transmissibility, coupling, and dynamic balance.

c. Other system configurations are then described. The two basic approaches to shock isolation, individual and group mounting, are considered. Some of the common ways of supporting a shock-isolated system are discussed, together with the factors that affect the selection of a particular isolator arrangement.

d. Actual shock isolation systems differ in many respects from simple theoretical systems. Actual systems are found to have distributed masses, stiffnesses and damping. Real transmissibility curves and transfer functions for such systems are presented to illustrate how they differ from those of idealized systems.

5-2. Requirement for isolation.

a. Comparison of the shock environment and equipment-hardness/personnel-tolerance will indicate the risk of shock damage. To the degree to which the shock environment exceeds the hardness/tolerance, shock isolation may or may not be required. The comparison can be made using shock spectra or Fourier magnitude spectra.

b. If the shock input does not exceed the hardness/fragility of the equipment or the personnel tolerance, the equipment can be hard-mounted to the structure and personnel will need no protection. This situation is illustrated in figure 5-1. The upper curve in each diagram shows the equipment hardness or fragility. The lower curve shows the facility environment. The figure illustrates the two forms of comparison of data in the frequency domain: (a) shock spectra, and (b) Fourier magnitude spectra.

c. If the shock input exceeds the equipment hardness or personnel tolerance, attenuation of the environment is required. This situation is illustrated in figure 5-2 for shock spectra and Fourier magnitude spectra.

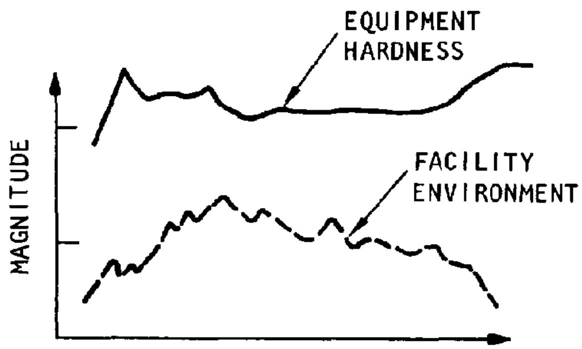
d. The insertion of an isolation system between equipment/personnel and the facility environment results in an alteration of the environment experienced by the equipment or personnel. This environment alteration, when applied to the environment of figure 5-2, may result in modified environmental curves such as those shown in figure 5-3.

e. The separation over the frequency range of equipment hardness/personnel tolerance and environment, such as illustrated in figures 5-1 and 5-3, is the desired goal of shock isolation. To accomplish this, shock-isolation systems can range from the basic spring-mounted system to servo-controlled systems. The range of isolation can be from moderate attenuation to extremely high.

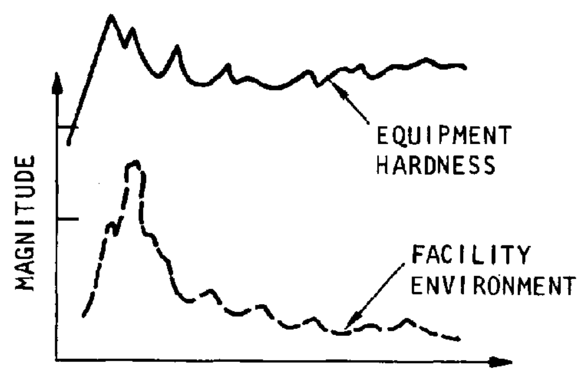
f. Unless equipment is specifically tested for a weapon system, information on equipment hardness may be minimal. Estimates of the environment of the facility or on the shock-isolated platforms must include the uncertainty of the threat and the uncertainty of the structural response prediction of the facility. These two uncertainties not only narrow the difference between the environment and equipment-hardness/personnel-tolerance, but can result in overlap in all or in some portions of the frequency spectrum.

g. Where the environment may exceed equipment hardness/personnel tolerances because of uncertainties, and where the problem cannot be resolved by equipment upgrading or shock isolation, a probabilistic approach may be used. A risk value, based on mission requirements of the weapon system, may be assigned in the form of probability of failure and confidence level. In a nominal case, this might be a 10 percent probability of failure with 90 percent confidence. In some cases, additional information may be required to generate more accurate failure probability and acceptable confidence levels.

h. Additional information may be required pertaining to equipment hardness and to the environment at the equipment location. The development of equipment hardness information is covered in chapter 4, which also describes methods for upgrading equipment. Additionally, physical models of the structure



(a) Shock spectra



(b) Fourier magnitude spectra

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Figure 5-1. Comparison of Facility Environment to Equipment Hardness Levels—No Isolation Required

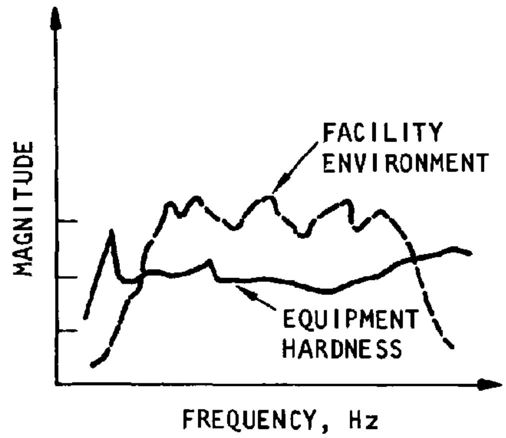
can be constructed for measurement of impedance and of mode shapes, and for high-explosive tests.

i. Cost/benefit studies for shock isolation can be used to determine the degree of isolation or the degree of environmental attenuation required. These studies should include the following considerations:

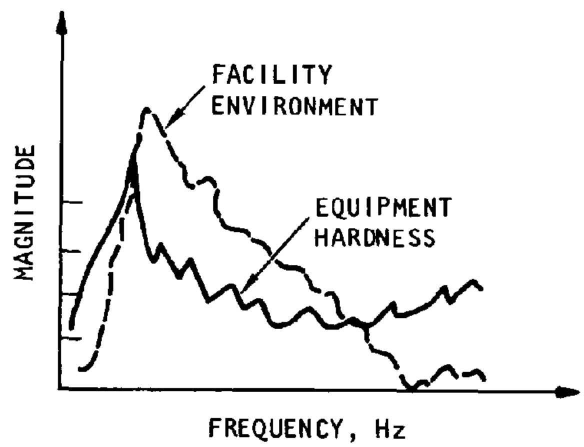
—System mission and its criticality
—Cost of refined environmental prediction
—Cost of equipment hardness upgrade
—Effect of personnel restraints (seated and prone) and personnel protection on operational effectiveness
—Impact of shock-isolation system on cost and size of protection facility
—Ratio of shock isolation cost to equipment hardness upgrading cost

5-3. Performance of single-degree-of-freedom systems.

a. In general, the analytical treatment of shock-isolation systems is based upon the principles of dynamic analysis. Since many isolated masses are a complex combination of numerous smaller components, each with its own static and dynamic characteristics, the first step in the analysis is to idealize the system. In most cases, this means representing the real system by a simplified model consisting of a rigid mass, or masses, connected by a spring and dash pot as shown in figure 5-4. This figure represents the simplest case, that of a single-degree-of-freedom system restrained to move in only one direction. In the more general case, the system would have at least six degrees of freedom,



(a) Shock spectra



(b) Fourier magnitude spectra

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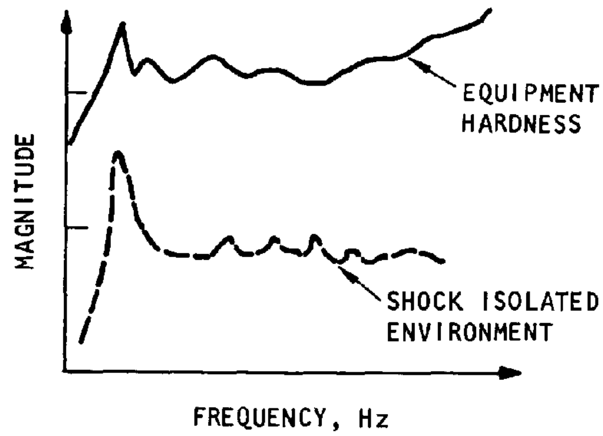
Figure 5-2. Comparison of Facility Environment to Equipment Hardness Levels—Isolation Required

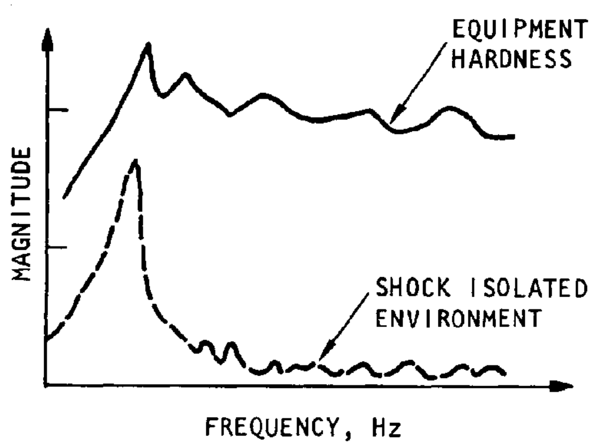
i.e., three displacements and three rotations. Under certain conditions, these six modes can be uncoupled and the system analyzed as six single-degree-of-freedom systems.

b. The single-degree-of-freedom system can also be used to illustrate the relation of some important parameters to the effectiveness of shock-isolation systems in general. In figure 5-4, the isolator is represented by the linear spring and viscous damping device enclosed within the dotted square. The suspended mass is assumed to be a rigid body. It is assumed that the base of the system is subjected to a periodic sinusoidal motion whose frequency is ω. The undamped natural frequency of the system is ω_N. The ratio of the maximum absolute displacement of the mass to the maximum displacement of the base is plotted in figure 5-5 as a function of the ratio of the frequency of the base motion to the natural frequency of the single-degree-of-freedom system. (The ratio of absolute velocities or of absolute accelerations of the base and of the mass can also be used, and are identical.)

c. The ratio of displacements is called the absolute transmissibility of the system. Curves are presented for various damping ratios (ratio of actual viscous damping coefficient to critical damping coefficient). When the frequency of the input motion is small compared to that of the system, the displacement of the mass is approximately equal to the displacement of the base. When the frequency of the base motion is several times that of the system, the motion of the mass is a



(a) Shock spectra



(b) Fourier magnitude spectra

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Figure 5-3. Effect of Shock Isolation in Attenuating Facility Environment

small fraction of the base motion. When the ratio of frequencies becomes large (e.g., 20 to 30), the system cannot respond to the base motions to any significant degree, as the mass tends to remain stationary in inertial space. At frequency ratios near one, large motions of the mass are possible and the magnitude is strongly affected by the amount of damping in the system. Here, the stiffness (spring) vector is equal and opposite to the mass vector, and the only limit on resonance motion is the energy dissipating damper.

d. One approach apparent from figure 5-5 is to use a low-frequency suspension system so that the ratio of frequencies is always large. However, low-frequency (or soft) systems have larger static and dynamic displacements and a greater probability of coupling between modes of vibration. Although soft systems may be acceptable under some conditions, other constraints will often prevent their use. One obvious constraint is a limit on the relative motion between the suspended mass and its supports or adjacent parts of the facility. This relative motion determines the amount of rattlespace that must be provided to avoid impact between the mass and other fixed or moving parts of the facility.

e. Figure 5-6 shows the variation of relative transmissibility with frequency ratio for several percentages of critical damping. Relative transmissibility is defined as the ratio of the maximum relative motion between the mass and base to the maximum displacement of the base. At low-frequency ratios (high fre-

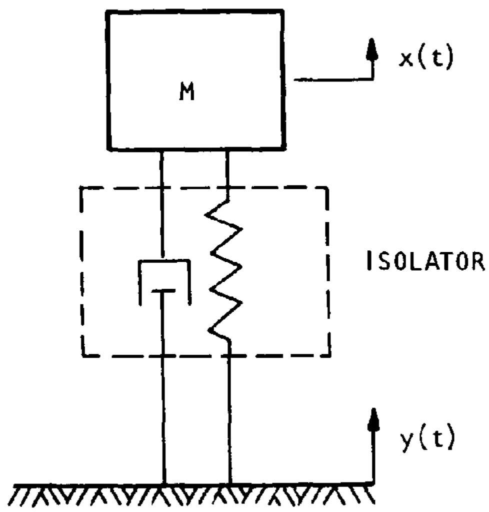


Figure 5-4. Idealized Model of Shock-Isolated Mass

quency, or hard, system), there is very little relative motion between mass and base. At high-frequency ratios (soft system), the relative displacement can be approximately equal to the base displacement. The damping ratio again strongly affects the response at frequency ratios near unity.

f. The acceleration of the mass is a function of the force applied to the mass by the spring and damping devices. In the case of linear undamped springs, the force is a function of the relative displacement between the mass and its support. In viscous damping devices, the damping force is a function of the percent of damping and the relative velocity between the mass and its supports. Thus, acceleration limits for the critical items will also impose restraints on spring stiffness and the amount of damping in the isolation system. In practice, the characteristics of the shock-isolation system are usually some compromise combination of spring stiffness and damping ratio to minimize input motions to the mass for a specified allowable rattlespace or to minimize rattlespace for specified allowable motions of the mass.

g. Figures 5-5 and 5-6 also demonstrate the need to avoid resonance (frequency ratio of one) between the system and disturbing function. Although the structural motions resulting from nuclear detonations are not steady-state sinusoidal in nature, they frequently are of an oscillatory type and the displacement-frequency ratio relationships discussed above are approximately applicable. AJA (1966) and Veletsos (1964) contain a more detailed discussion of the effects of load duration, nonlinear springs, damping, and system frequency on response.

h. In the general case, a single-mass system can have six degrees of freedom: translation in three orthogonal directions and rotations about three orthogonal axes. These systems can also be classified as coupled or uncoupled. A coupled system is one in which forces or displacements in one mode (or direction) will affect or cause response in another mode, e.g., a vertical displacement of a single rigid mass might cause

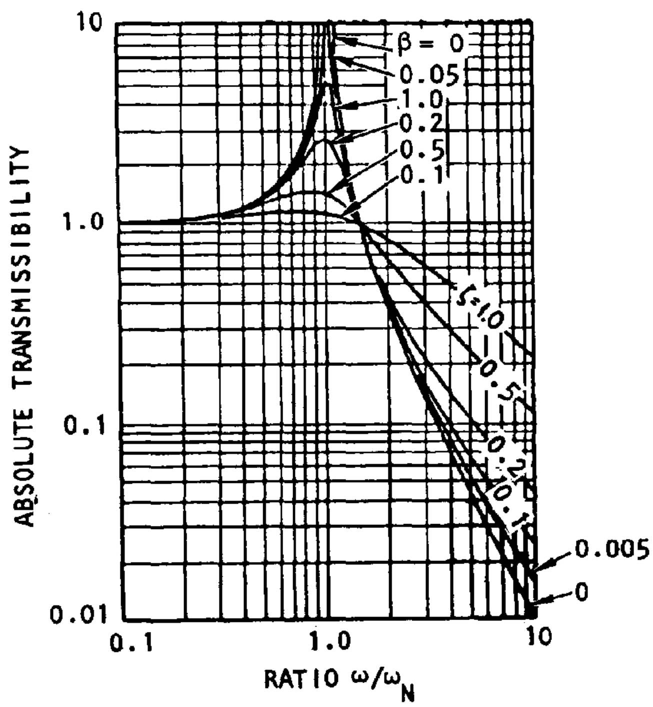


Figure 5-5. Effect of Frequency Ratio and Damping Ratio on Absolute Transmissibility (Harris-Crede, 1976)

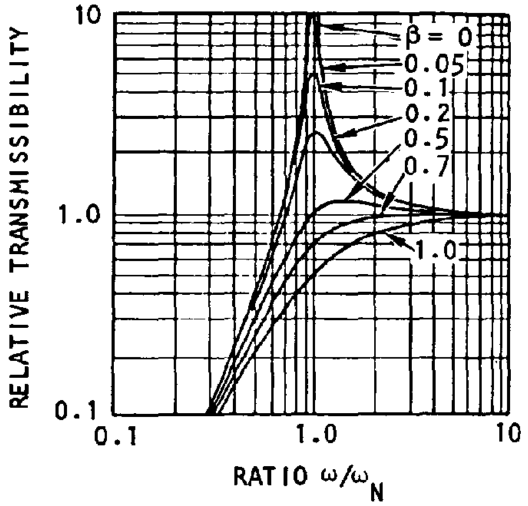


Figure 5-6. Effect of Frequency Ratio and Damping Ratio on Relative Transmissibility (Harris-Crede, 1976)

rotation of the mass about some axis. An uncoupled system, on the other hand, is one where forces or displacements in one mode do not generate a response in another mode. If the system is completely uncoupled, base translations in any one of three orthogonal directions will excite translations of the mass only in that direction. Similarly, a pure rotation of the base about any one of three orthogonal principal inertia axes through the mass center will excite only pure rotations of the body about that axis. The principal inertia axes are those about which the products of inertia vanish. The principal elastic axes of a resilient element (isolator) are those axes for which an unconstrained element will experience a displacement colinear with the direction of the applied force. The point of intersection of the principal elastic axes of a resilient element is called the elastic center of the resilient element. If the principal elastic axes and the principal inertia axes of the shock-isolated system coincide, the modes of vibration are uncoupled. The origin or point of intersection of both sets of axes must lie at the center of gravity of the mass. Such a system is also referred to as a balanced system.

i. In figure 5-7, if all the springs have the same stiffness, the elastic center will be located at point A. If the suspended block is of uniform density, its center of gravity is also located at A, and the system is uncoupled for motions input through the springs. Some systems may be uncoupled only for motions in a particular direction. If point B in figure 5-7 is the center of gravity of the mass, a pure horizontal displacement of the structure would excite only a horizontal displacement of the mass. A pure vertical displacement of the structure would excite both vertical and rotational displacements of the mass. Thus, vertical and rotational modes are coupled. If the center of gravity were located at point C, then horizontal, vertical, and rotational modes would be coupled.

j. If the characteristics of the mass and shock-isolation system are such that the modes of vibration can be uncoupled, the system can be analyzed as a series of independent single-degree-of-freedom systems. The re-

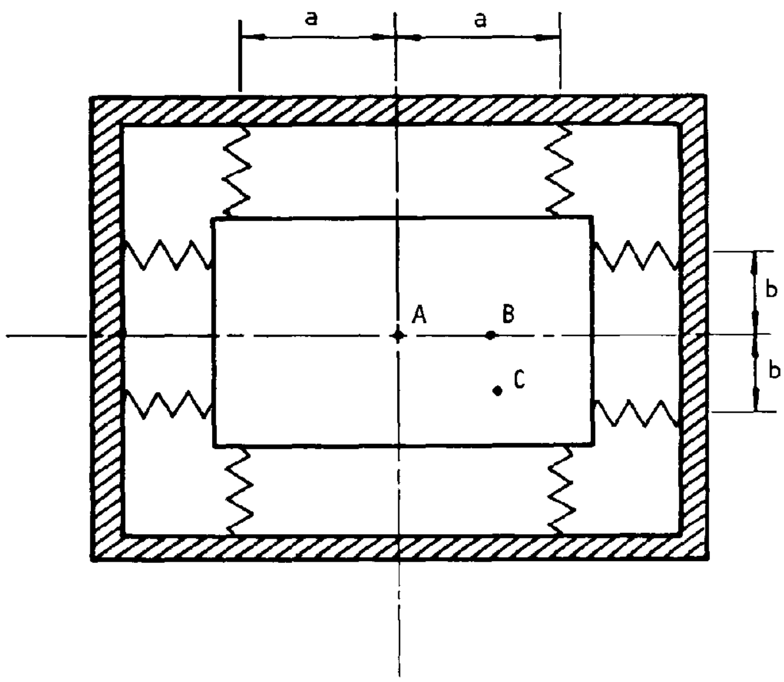


Figure 5-7. Shock Isolation System (Crawford et al., 1974)

sponse of each of these systems can be determined on the basis of input motions and isolator properties in a direction parallel to or about one of the principal inertia axes. The response in each one of these modes can be summed in various ways to obtain the total response of the system. The sum of the maximum responses would neglect differences in phasing and should represent an upper limit of the actual motions. Since it is unlikely that maximum response will occur simultaneously in all modes, the square root of the sum of the squares of the maximums (root sum square values) may represent a more realistic maximum. Superposition of modal responses is appropriate, of course, only for elastic systems.

k. Although it is not always possible to achieve a dynamically balanced shock-isolation system, it offers advantages other than a simplification of the computation effort. Saffell (1971) suggests that a dynamically balanced system will also result in reduced motions during oscillation. Since modes are not coupled in a balanced system and rotational inputs to the system are usually of small magnitude in protective construction applications, rotational motions of the suspended mass will be minimized. This is particularly important in the case of large masses, where small angles of rotation can result in large translations at extreme locations far from the center of gravity.

l. Because of the advantages of a dynamically balanced system, various approaches are sometimes taken to minimize coupling of modes. One criterion is that frequencies in the six modes should be separated sufficiently to avoid resonance between the modes. This separation should be maintained over some reasonable range of loads that the system might be expected to sustain during its lifetime. Because of the importance of minimizing rotational modes of response, Saffell (1971) also suggests avoiding extremely low stiffnesses in these modes. Although increased stiffnesses in the rotational modes will add higher frequency components to system response, there is little rotational energy introduced directly into them, if the system is properly balanced. Higher rotational stiffnesses will also minimize residual inclination of the isolated platform under changing static loads or due to friction forces in the isolators. Most isolation systems will incorporate some features that will allow for changes in the total load or distribution of load on the isolation platform. The adjustments can be made by changing properties of the isolators themselves or adjustment of ballast weights on the platform.

m. If the dynamic system is also a multiple mass system, the system can be analyzed by the modal method of analysis or one of the numerical integration techniques. The modal method of analysis requires solution of simultaneous equations of motion to determine characteristics shapes and frequencies of each mode and is limited to the elastic case. The numerical techniques do not require prediction of mode shapes and frequencies and will handle both elastic and inelastic response. The modal method of analysis or numerical integration techniques can also be used for the analysis of multiple-degree-of-freedom, single-mass systems where the various modes of response are coupled.

5-4. Performance of other system configurations.

a. Individual versus group mounting.

(1) The two basic approaches to shock isolation in protective construction are to provide individually tailored systems for each component or to group together two or more items on a common platform. In the latter case, a system is selected to satisfy the most stringent criteria. In some cases, where the shock tolerance of the various items differs greatly, a combination of the two approaches may be the most effective solution. Although the relative location or size of some items may make individual mounts the more practical approach in certain cases, group mounting will generally be as reliable and the least costly solution.

(2) There are several advantages of group-mounted systems. A group-mounted system will be less sensitive to variations in weights of individual items of equipment because of the larger combined weight of all items and the platform. With a number of items, there is also a greater flexibility in controlling the center of gravity of the total mass. A group-mounted system can also require less rattlespace than several independently mounted systems; furthermore, interconnections between components is greatly simplified if they are all mounted on a single platform.

(3) An important advantage of group systems is cost. Individual mounts require a large number of isolator units with adjustment systems to accommodate changes in weights or performance requirements. If adjustment capability is not provided, the isolators may have to be redesigned or replaced if weight and performance characteristics change significantly. Although larger, more costly units are required for the group-mounted system, fewer numbers are required, and the cost per pound of supported load can be much lower. The maintenance and spares cost should also be less for the fewer number and types of units required in the group-mounted system.

b. Platform characteristics. It is desirable that the platform for group-mounted systems be sufficiently stiff so that the platform and associated mounted equipment can be treated as a rigid body. This criterion is usually satisfied if the lowest natural frequency of the platform is several times larger than the natural frequency of the spring-mass system. When large heavy items of equipment are involved, platforms meeting this stiffness criterion may not be practical. In such cases, it will be necessary to treat the platform-equipment configuration as a multimass system.

c. *Isolator arrangements.*

(1) There are many ways to support a shock-isolated item. Some desirable features have been mentioned previously in connection with dynamically balanced systems. Some of the more important factors affecting the selection of an isolator arrangement are

- —The site, weight, shape, and location of the center of gravity of the suspended mass must be determined
- —The direction and magnitude of the input motions must be determined
- —Rotation of the lines of action of the devices should be small over the full range of displacements of the system to avoid system nonlinearities
- —The coupling of modes should be minimized
- —Static and dynamic instability must be prevented
- —It is desirable in most cases, and necessary in some, that the system return to its nominal position
- —The space available for the isolation system must be determined
- —The type of isolation devices to be used must be considered

(2) Some of the more common general arrangements of isolators are shown in figures 5-8, 5-9, 5-10, and 5-11. All systems shown are assumed to have the

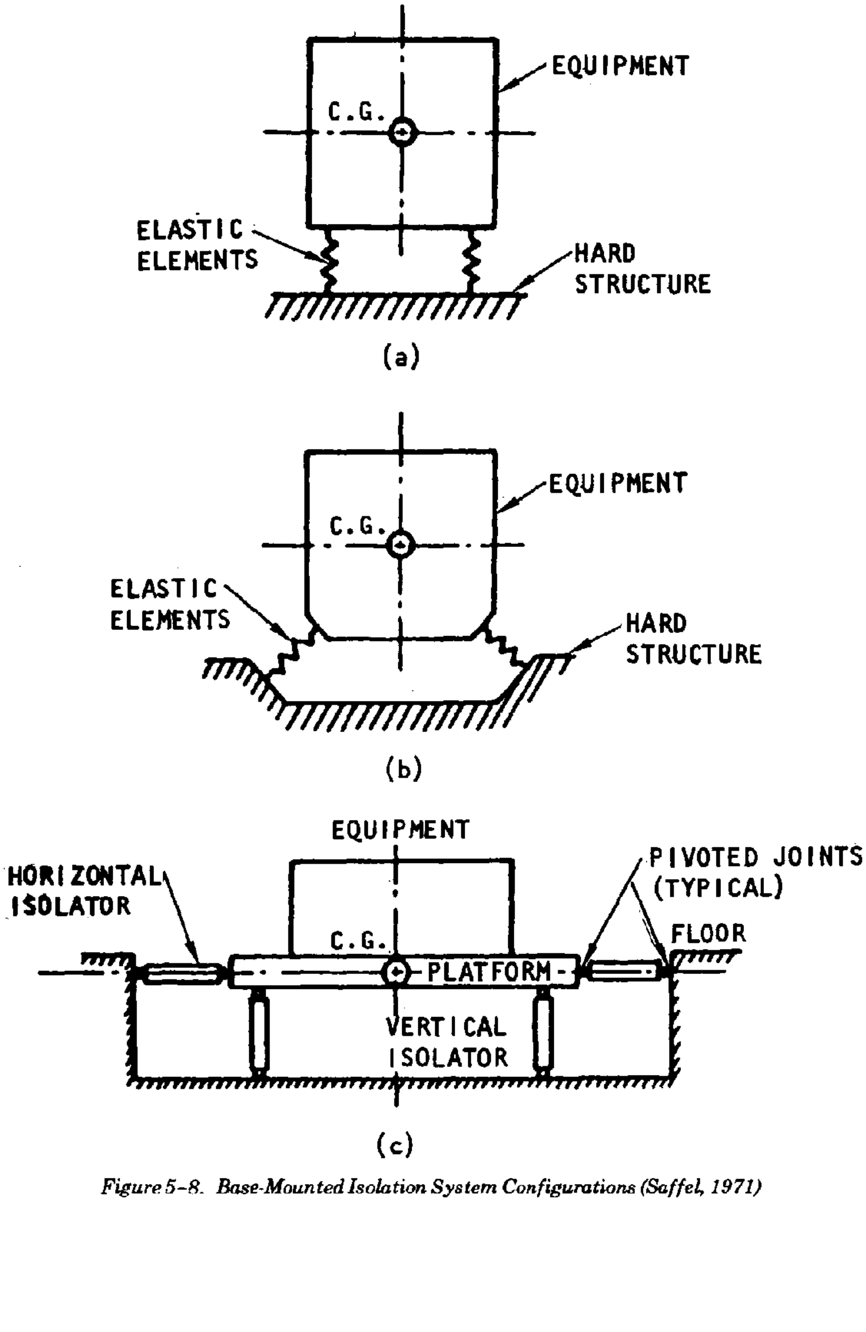


Figure 5-8. Base-Mounted Isolation System Configurations (Saffel, 1971)

same arrangement of isolators in a plane through the center of gravity (c.g.) and perpendicular to the surface of the page.

(3) A symmetrical system would be one in which the elastic axes and principal inertia axes intersect at a point. Since the elastic axes of the suspension system and the principal inertia axes of the mass coincide, the system is dynamically balanced and all modes are uncoupled.

(4) In figure 5-8a, the mass is supported by four vertical isolators. These isolators must provide horizontal, vertical, and rotational stiffnesses in order for the system to be stable under all possible motions. There will be coupling between horizontal displacements and rotations about horizontal axes. This arrangement and that shown in figure 5-8b are appropriate for those applications where there are no convenient supports for horizontal isolators. Figure 5-8b is a preferred arrangement, since the line of action of the isolators can be directed towards the c.g. of the mass to allow decoupling of some modes. As in the case of figure 5-8a, the isolator elements must have adequate stiffness in axial and lateral directions to ensure stability under static and dynamic conditions. In figure 5-8c the isolator elements are oriented parallel to the three orthogonal system axes. This arrangement provides system stability even when the isolator elements have only axial stiffness. If the c.g. of the suspended mass is located as shown, decoupling of modes is possible. While it is possible to select isolator properties and geometries so that the lines of action of the isolators pass through the c.g. of the system under static conditions, response of the system to base motions will obviously alter its geometry. When the line of action of the isolators is changed due to displacement of the mass relative to its supports, coupling of modes of vibration will be introduced. The degree of coupling will be affected by the magnitude of the dis-

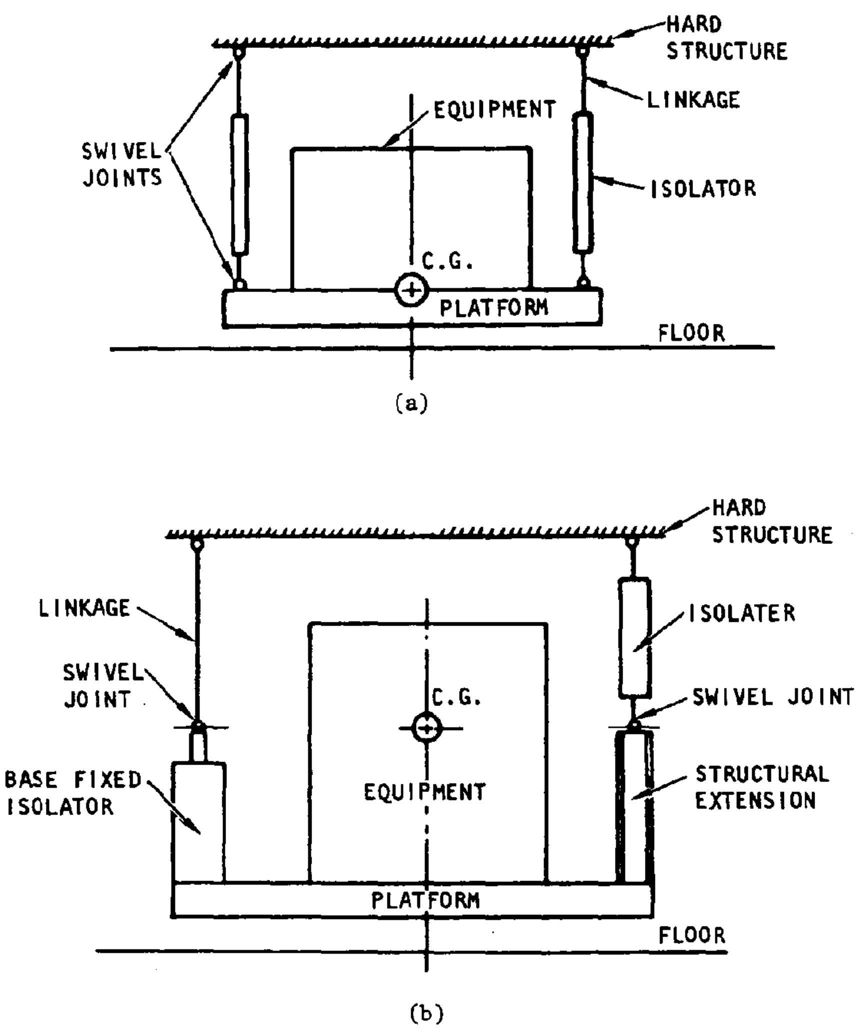


Figure 5-9. Overhead Pendulum Shock Isolation Systems Using Platforms (Saffel, 1971)

placements and the length of the isolators. Although a system which would provide complete decoupling of modes of vibration under all conditions of response is probably impractical, isolator properties and arrangements should be selected so as to minimize the effect of displacements.

(5) Figure 5-9 shows two arrangements of overhead pendulum shock isolation devices using platforms to support the sensitive components. In both cases the c.g. of the suspended mass is relatively low. These types of suspension systems have been used extensively for supporting large loads in protective structures. The overhead pendulum system normally uses swivel joints at the points of attachment of the isolator elements, so gravity provides the horizontal restoring force (or stiffness). This force and the frequency of the system in the horizontal mode are a function of the total weight of the suspended mass and the length of the pendulum. Each pendulum arm also includes an isolator element in series which determines system stiffness in the vertical direction. The system is linear for small angular displacements, i.e., $\theta \cong \sin \theta$. Insertion of the vertical isolator elements can introduce additional nonlinearities and coupling between the pendulum and vertical spring modes. Parsons (1962) suggests that if the uncoupled pendulum frequency is near one-half the uncoupled vertical spring frequency, interchange of energy between the modes can lead to pendulum motions greatly exceeding those predicted by linear assumptions. A detailed discussion of this problem is presented in Parsons (1962), Sevin (1960), and Seven (1961).

(6) Since most pendulum systems have low natural frequencies, they are displacement sensitive and will normally require greater rattlespace than other systems. They exhibit very little damping in horizontal modes, and it is frequently necessary to add some type of sway dampers as shown in figure 5-10a. These dampers can be one of several types including gas, liquid or friction systems. Since one of the main advantages of overhead pendulum systems is that they require no horizontal stiffness elements, their attractiveness is greatly diminished in those cases requiring horizontal damping.

(7) The point of attachment of the isolator swivel joint to the platform determines the location of the horizontal elastic axis of the system. Figure 5-9b shows two ways of varying the point of attachment so that the horizontal elastic axis can be made to coincide with the c.g. of the suspended mass at the equilibrium position and help minimize coupling between the modes of response.

(8) Another solution to the low horizontal stiffness and damping characteristics of the overhead pendulum system is to incline the pendulum arms as shown in figure 5-10b. Saffell (1964) presents a detailed analysis of both configurations and the factors affecting their vibration characteristics. It concludes that the inclined pendulum system is less sensitive to changes in c.g., provides static and dynamic stability over a wider range of displacements and loading conditions, minimizes coupling between modes, and provides rapid restoration of the system to an equilibrium position.

(9) Another type of overhead pendulum system is shown in figure 5-11. This type has been used for missile suspension systems in silos. The pendulum arms can be vertical or inclined. In this system the base dimension may be less than one-half the vertical distance from the platform to the c.g. of the mass. Since the stiffness of the isolators is limited by vertical isolation requirements, static and dynamic stability problems can arise, even with small rotations of the platform.

(10) Table 8-1 summarizes pertinent characteristics of the general types of isolation systems. The fre-

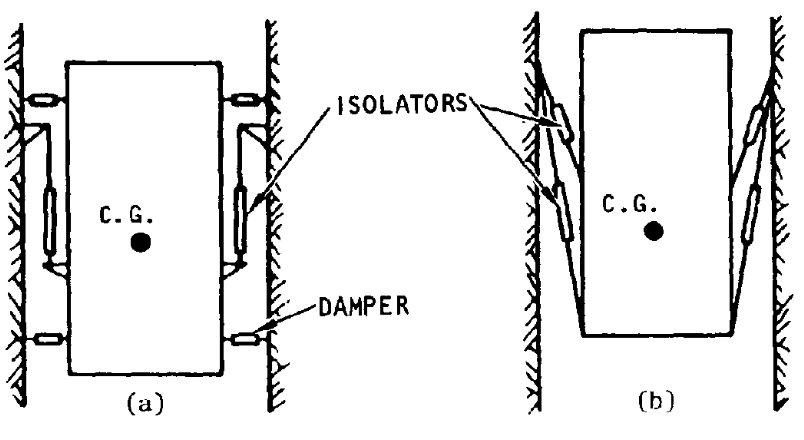


Figure 5-10. Variations of Simple Pendulum Systems (Saffel, 1964)

quency ranges and relative displacements referred to in table 8-1 are defined as follows:

Low-frequency range	Less than 1 Hz
Mid-frequency range	1 to 10 Hz
Large relative displacements	Less than 40 in. (102 cm) vertical and 14 in. (35.6 cm) horizontal (combined)
Moderate relative displacements	Less than 15 in. (38.1 cm) vertical and 5 in. (12.7 cm) horizontal (combined)
Small relative displacements	Less than 6 in (15.2 cm) vertical and 2 in. (5.08 cm) horizontal (combined)

(11) The comment in table 8-1 regarding nonlinearity refers to the error in assuming linear behavior when actual response is nonlinear. The systems shown in figure 5-8 can be considered equivalent to column (1) of table 8-1 and those of figure 5-9 to column (4).

5-5. Performance of actual shock isolation systems.

a. In the foregoing description of shock-isolation system performance, the mass, spring, and damper elements are ideal: the mass contains no damping or springs, the spring is linear and massless, and the damper is purely viscous without spring rate or mass.

b. In practical cases, the isolated platform and mounted equipment are distributed systems of masses, stiffnesses, and damping. The damping in structures is complicated by the various mechanisms in which it is manifested, such as friction, hysteresis, aerodynamic effects, and material. Isolators vary with type and design and depart from ideal springs, as they include damping and mass characteristics which modify their performance. The elastic isolator axes and mass center of gravity seldom coincide; therefore the normal condition is a coupled system. Large platforms with heavy equipment often have platform bending or torsion in the neighborhood of the "rigid body" resonant frequency of the platform.

c. Transmissibility curves or transfer functions provide a means of measuring performance of an isolation system. The curves of figure 5-5 for an ideal isolation system may be compared to those obtained with actual physical systems. Transfer functions for small isolation systems are routinely measured for aircraft, missile, and automotive systems. Only recently have measurements become available for large shock-isolated platforms, such as those located in the Safeguard ABM system (Safford-Walker, 1975).

d. These measured transfer functions, as would be expected, exhibit considerable deviation from the ideal system. Typical measurement data is presented in figure 5-12. This system weighed 198,000 lb and measured 64 ft by 49 ft. Eighteen helical coil springs supported the platform; damping was supplied by separate Coulomb friction dampers at each isolator location.

e. Figure 5-12 shows the transfer function magnitude and phase in the frequency domain, from 0.5 Hz to 500 Hz, and also the impulse function in the time

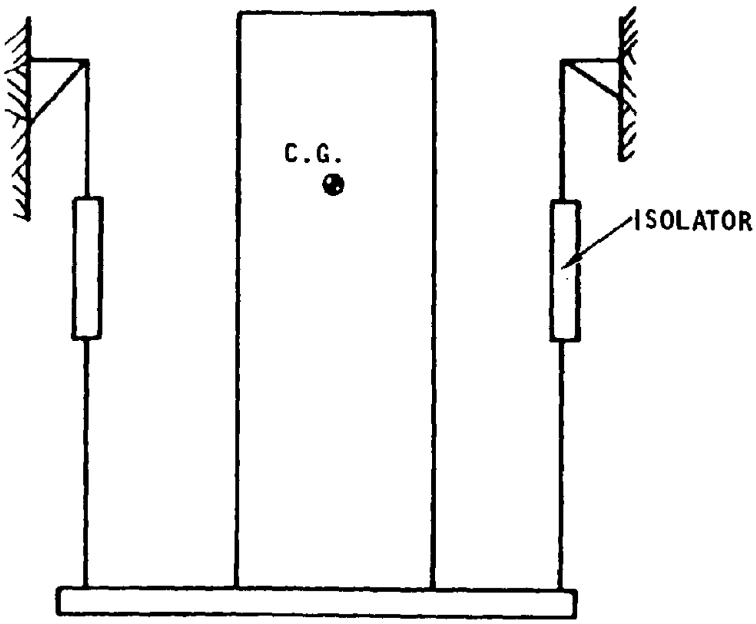


Figure 5-11. High Center of Gravity Pendulum Systems (AJA, 1966)

domain. "Rigid body" isolation resonance for this system was 1 Hz; however, major bending modes of the platform started at 0.6 Hz. Figure 5-12a shows amplification of input motion in the low frequencies of up to a factor of 2.25, with attenuation (below unity) starting at 9 Hz. At 17 Hz a surge (resonance) of the large spring isolators occurred. From 17 Hz to 500 Hz isolation attenuation of 70 percent or better was obtained. The various spikes in the figure are traceable to major resonances of the system and of the support equipment.

f. Figures 5-13 to 5-22 present transfer function magnitudes from 30 Hz to 500 Hz for conventional large shock-isolation systems. These measurements were made on the Safeguard ABM system (Safford-Walker, 1975). Isolators were helical coil springs, with and without external friction dampers, or pendulums with integral pneumatic springs. The pneumatic isolators included internal orifices for viscous damping.

g. Each figure contains a brief tabulation of the shock-isolation system and size of the platform. The weight of equipments and platforms ranged from 7500 lb to 244,000 lb; platform areas ranged from 48 sq ft to 3136 sq ft. Four isolators were used for the small platforms and up to 20 isolators for the larger sizes.

h. Review of the figures does not reveal a significant difference between helical coil spring isolators and pneumatic isolators. Performance characteristics in the frequency range measured were dominated by the structural dynamic properties of the platforms and equipments.

i. Figure 5-20 illustrates an effective isolation system, where both the platform and equipment were relatively rigid. Figures 5-21 and 5-22 present information for comparable switchgear equipment mounted on a platform. The improved performance in figure 5-22 over that in figure 5-21 was due to the concrete fill used in the platform for the walkway.

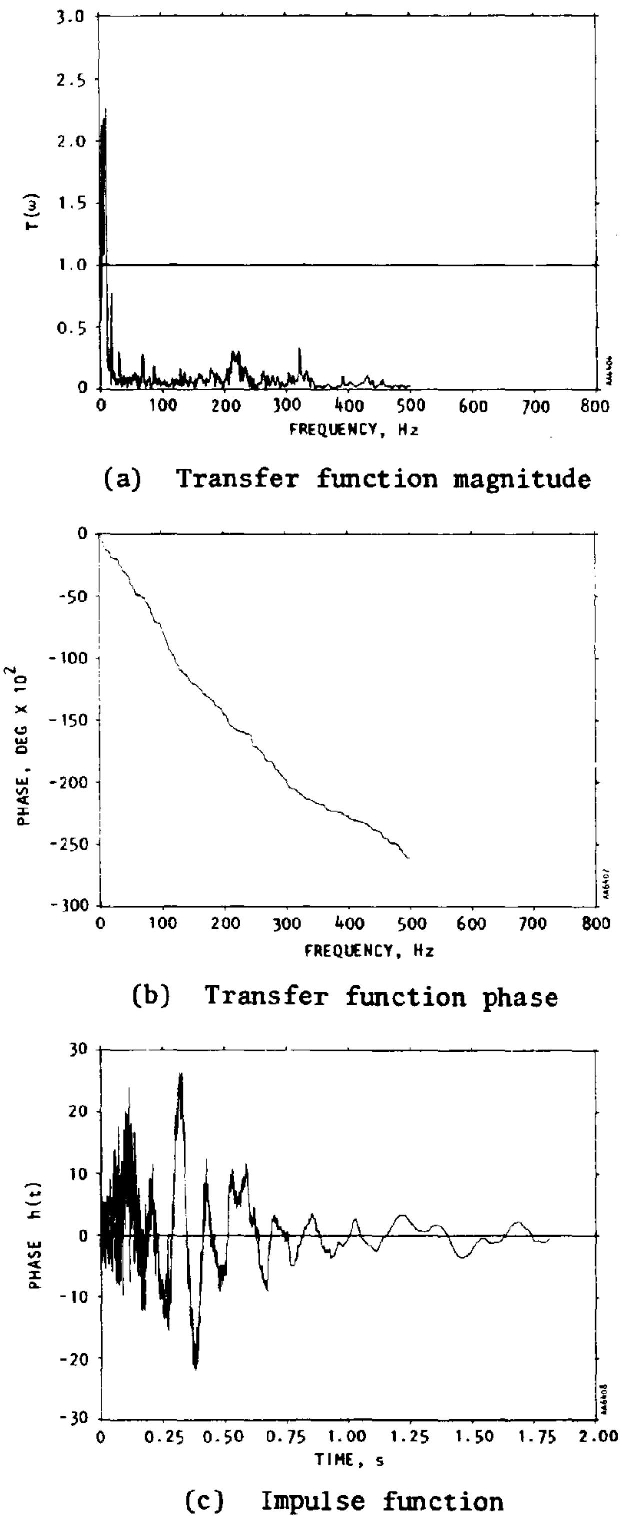


Figure 5-12. PARPP Control Room, Safeguard System: Measured Transfer Function for 0.5 Hz to 500 Hz (Safford-Walker, 1975)

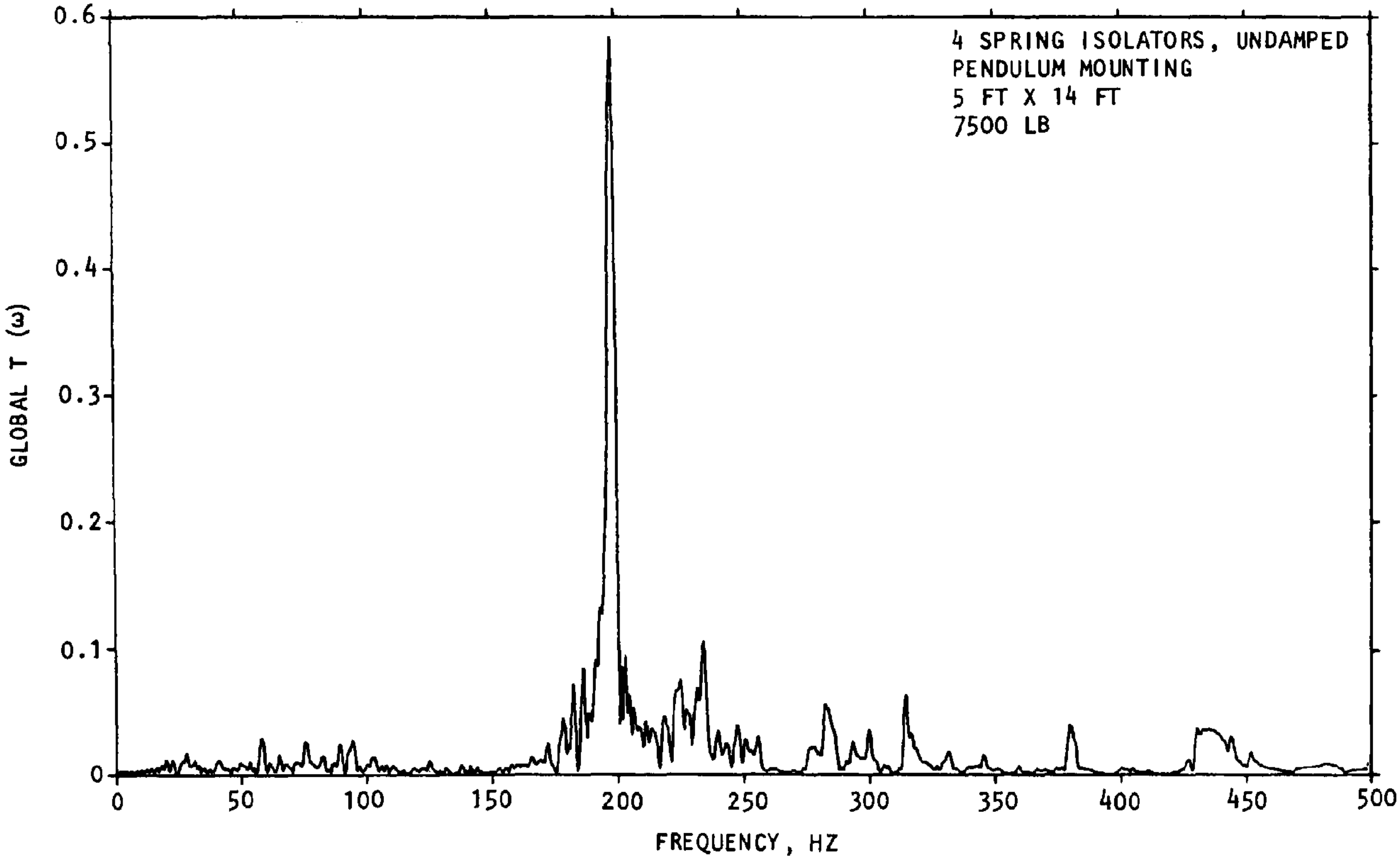


Figure 5-13. Platform PARPP-H: Transfer Function Modulus (Safford-Walker, 1975)

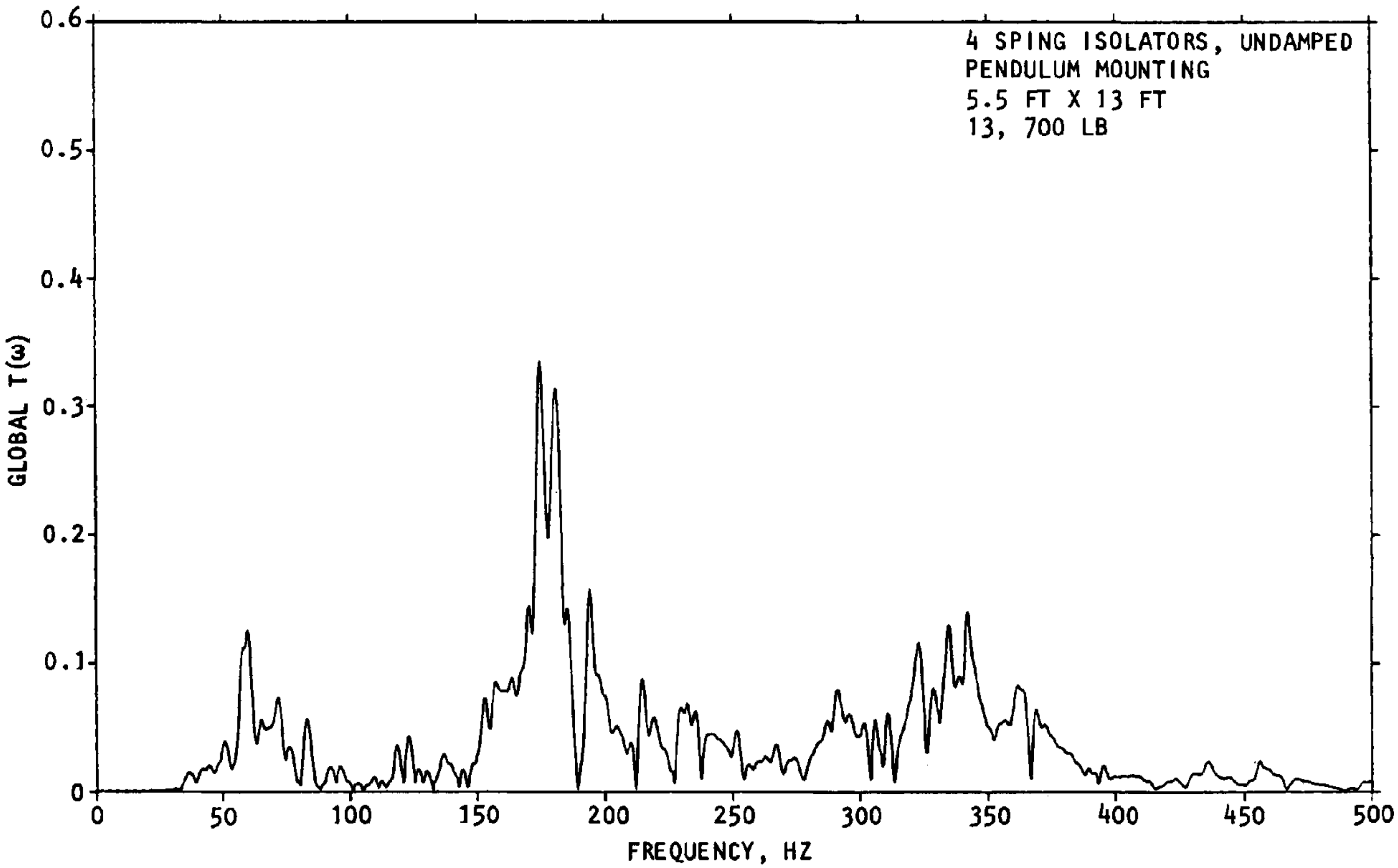


Figure 5-14. Platform PARPP-D: Transfer Function Modulus (Safford-Walker, 1975)

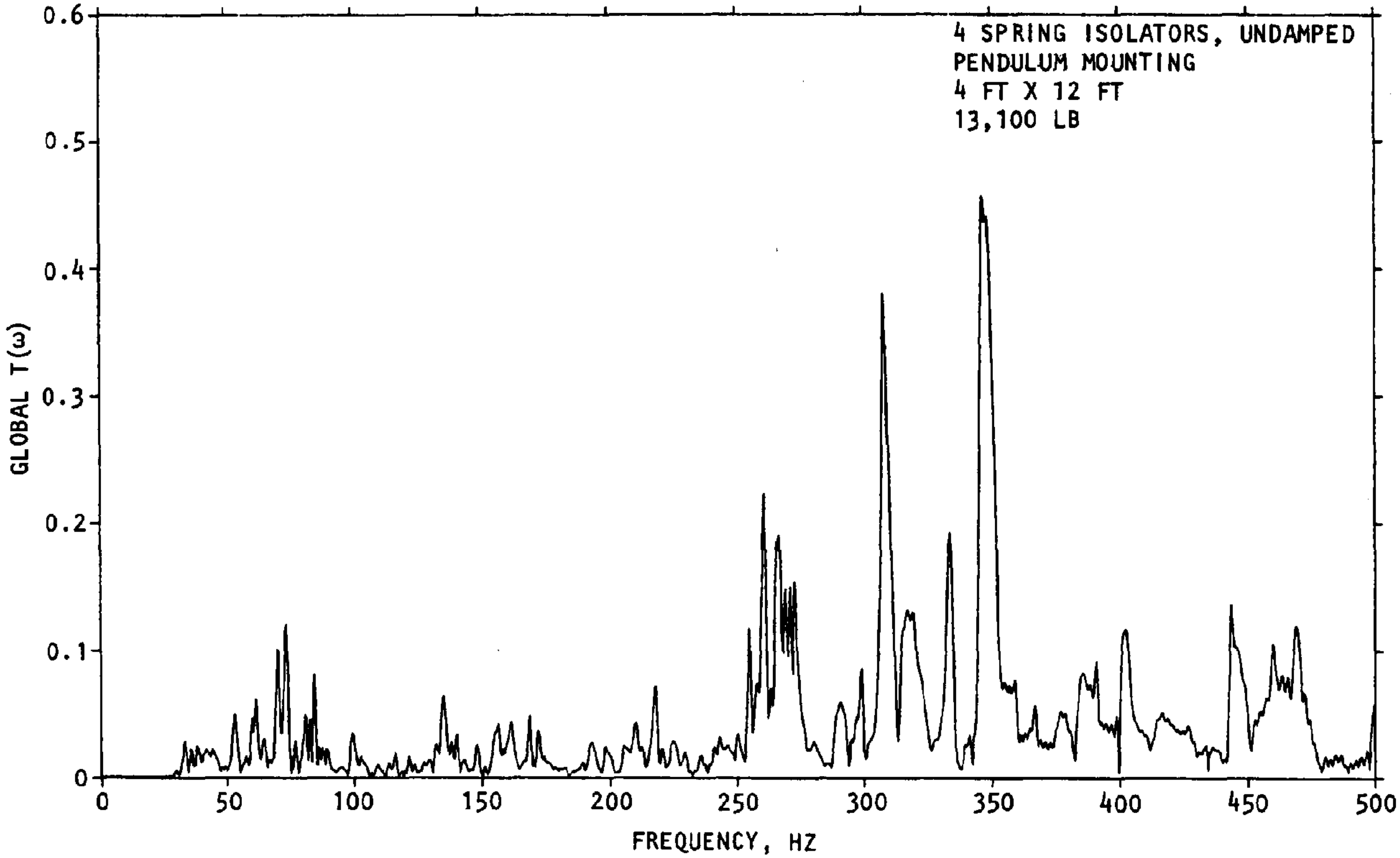


Figure 5-15. Platform PARPP-G: Transfer Function Modulus (Safford-Walker, 1975)

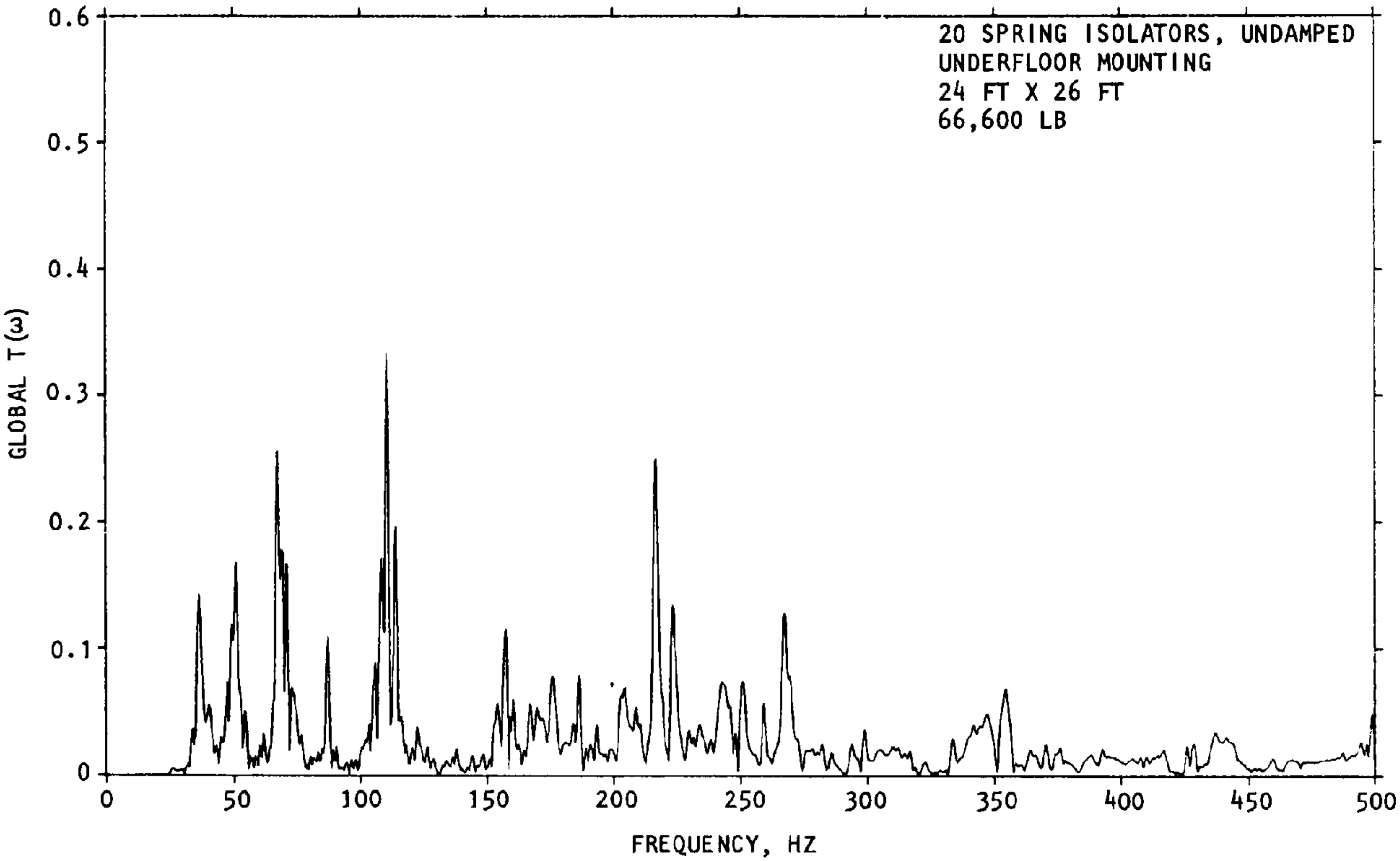


Figure 5-16. Platform MSRPP 9-10: Transfer Function Modulus (Safford-Walker, 1975)

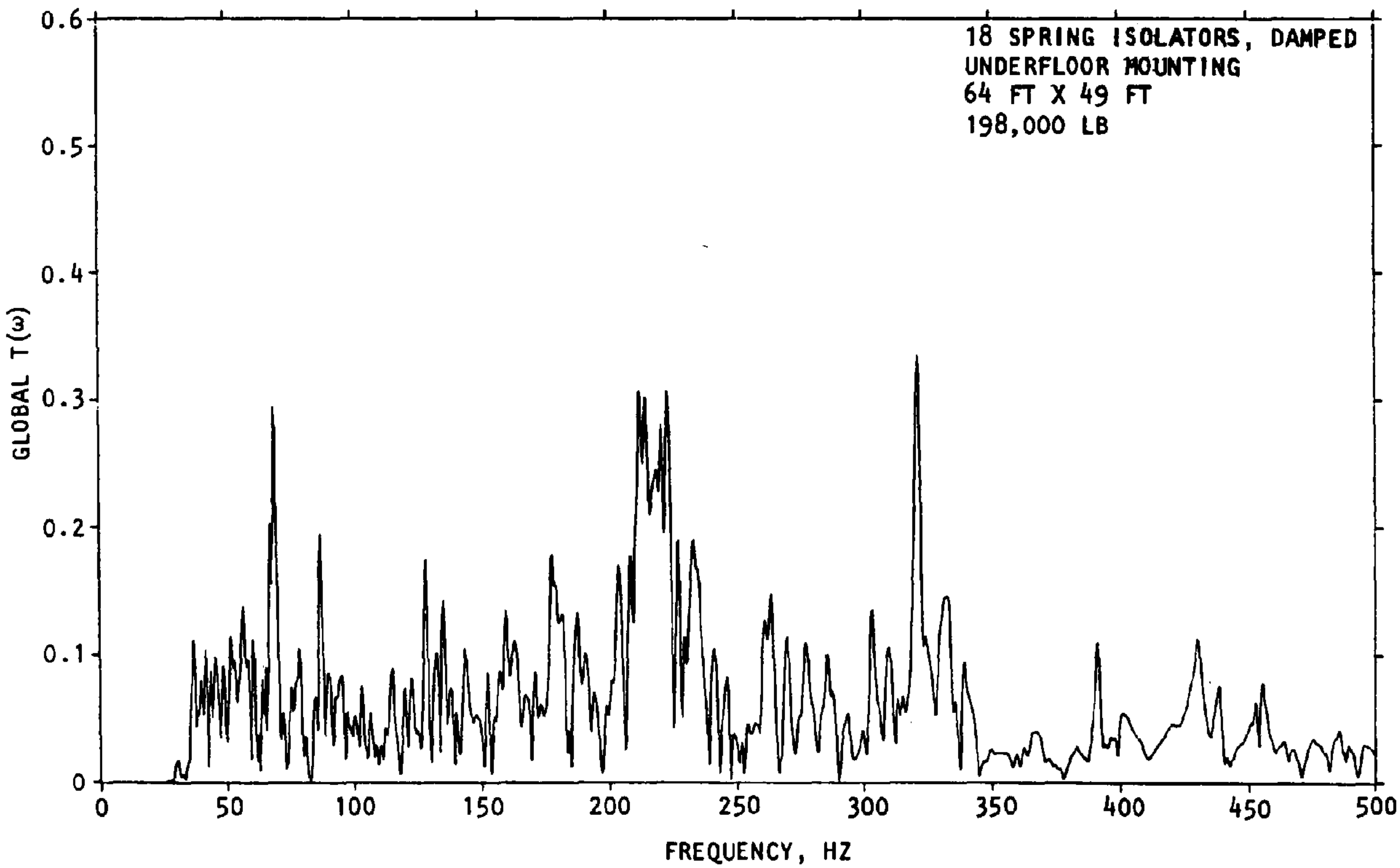


Figure 5-17. Platform PARPP-CR: Transfer Function Modulus (Safford-Walker, 1975)

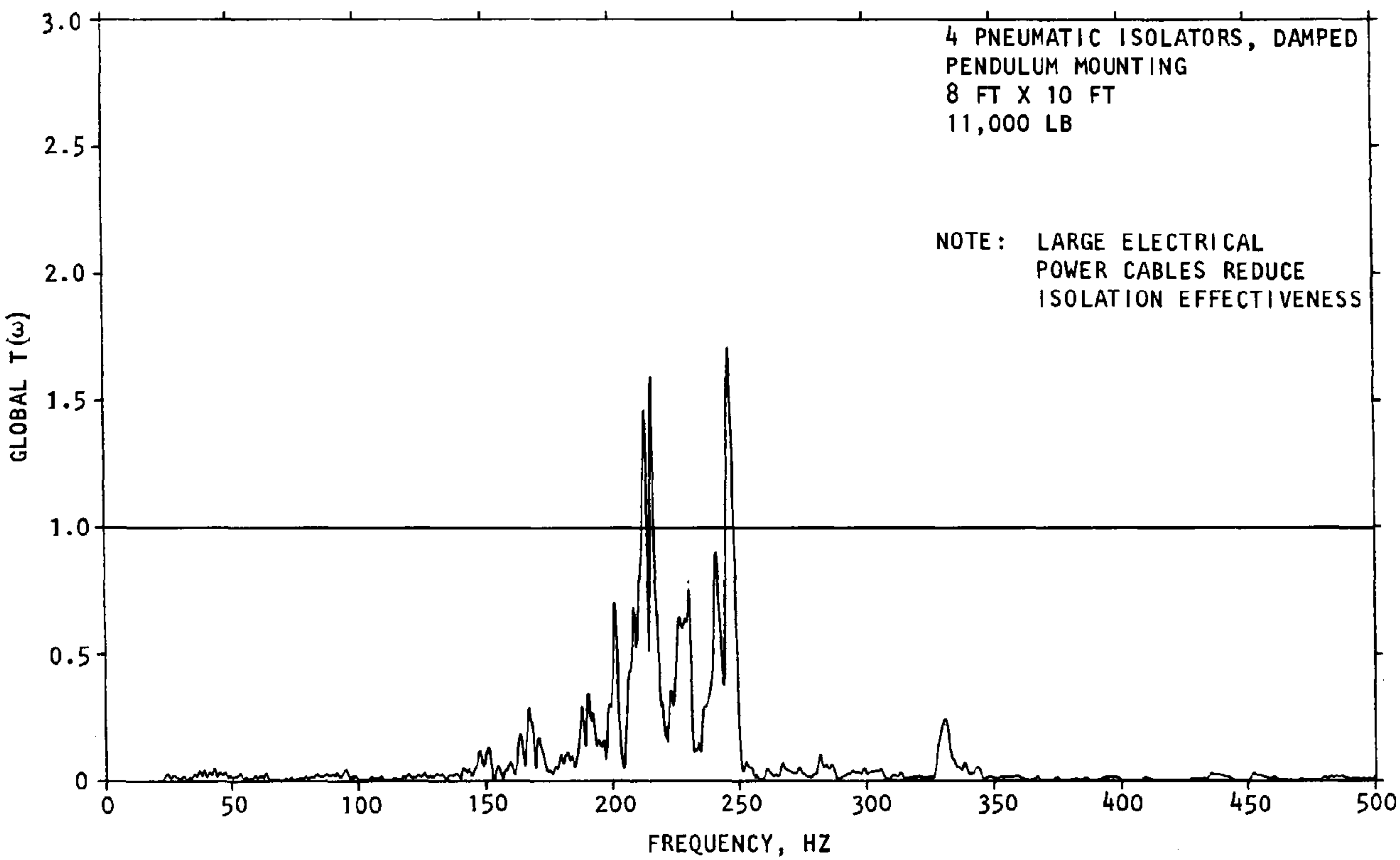


Figure 5-18. Platform RLOB-124B: Transfer Function Modulus (Safford-Walker, 1975)

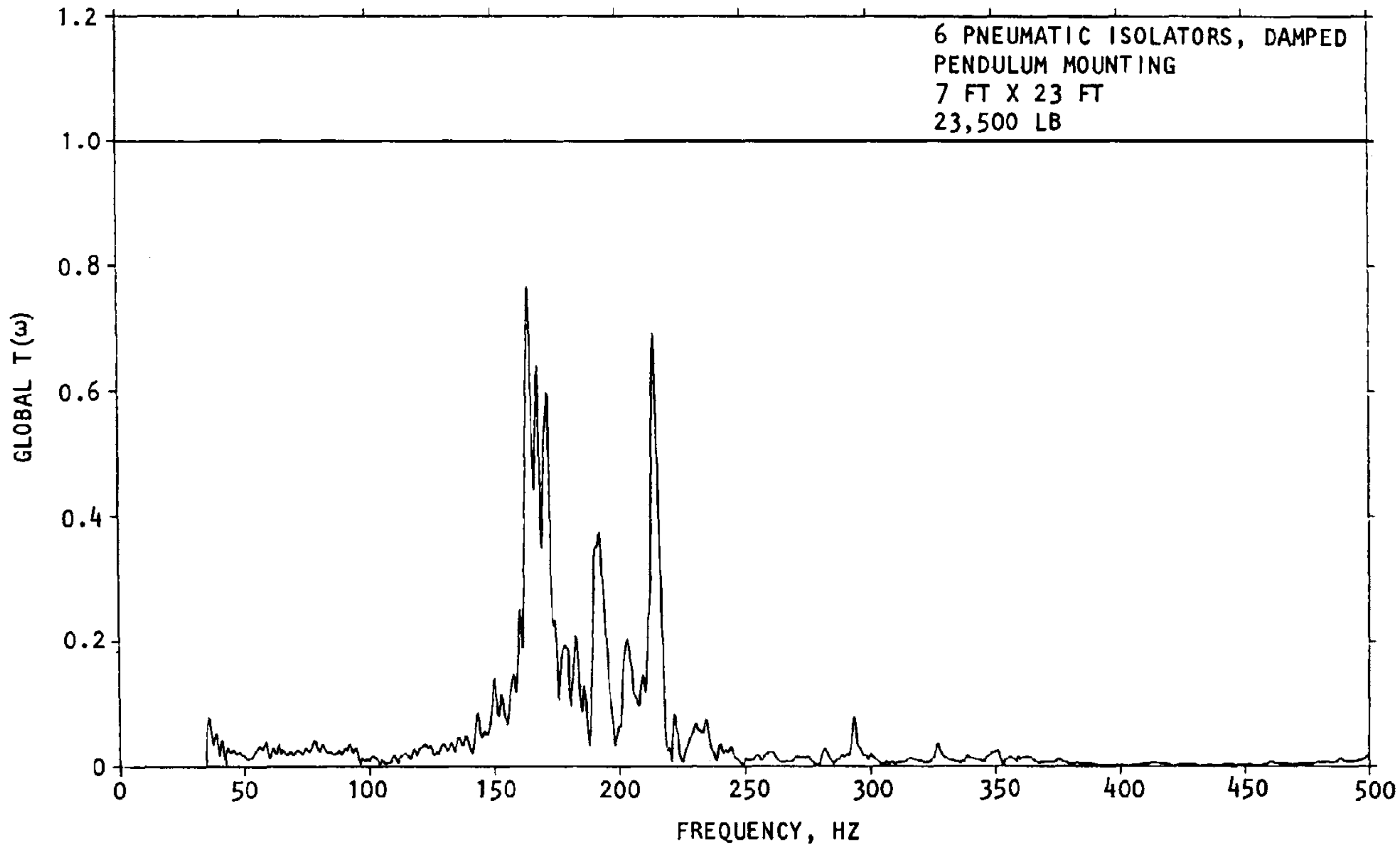


Figure 5-19. Platform RLOB-124C: Transfer Function Modulus (Safford-Walker, 1975)

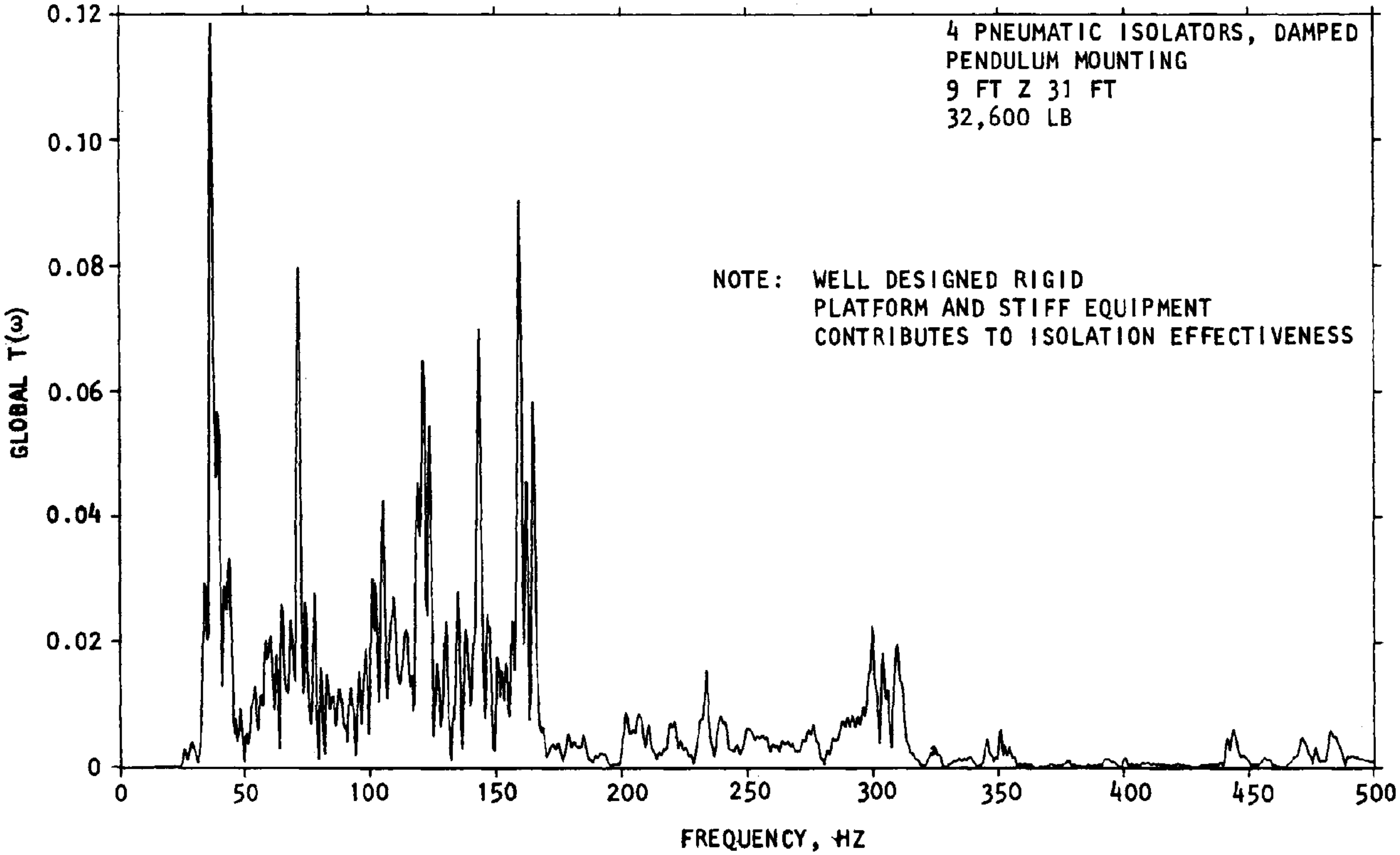


Figure 5-20. Platform MSCB-125B: Transfer Function Modulus (Safford-Walker, 1975)

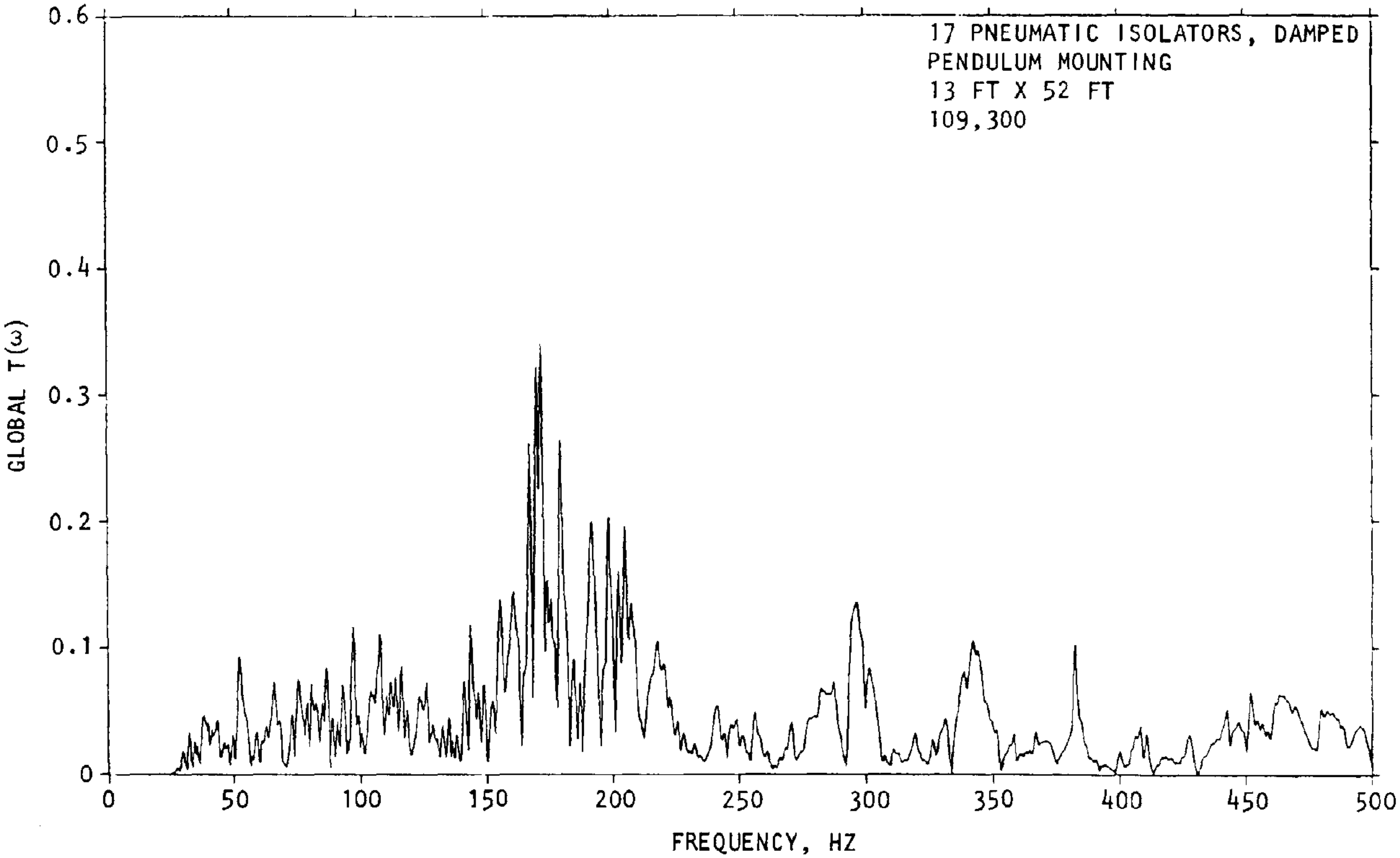


Figure 5-21. Platform PARPP-A: Transfer Function Modulus (Safford-Walker, 1975)

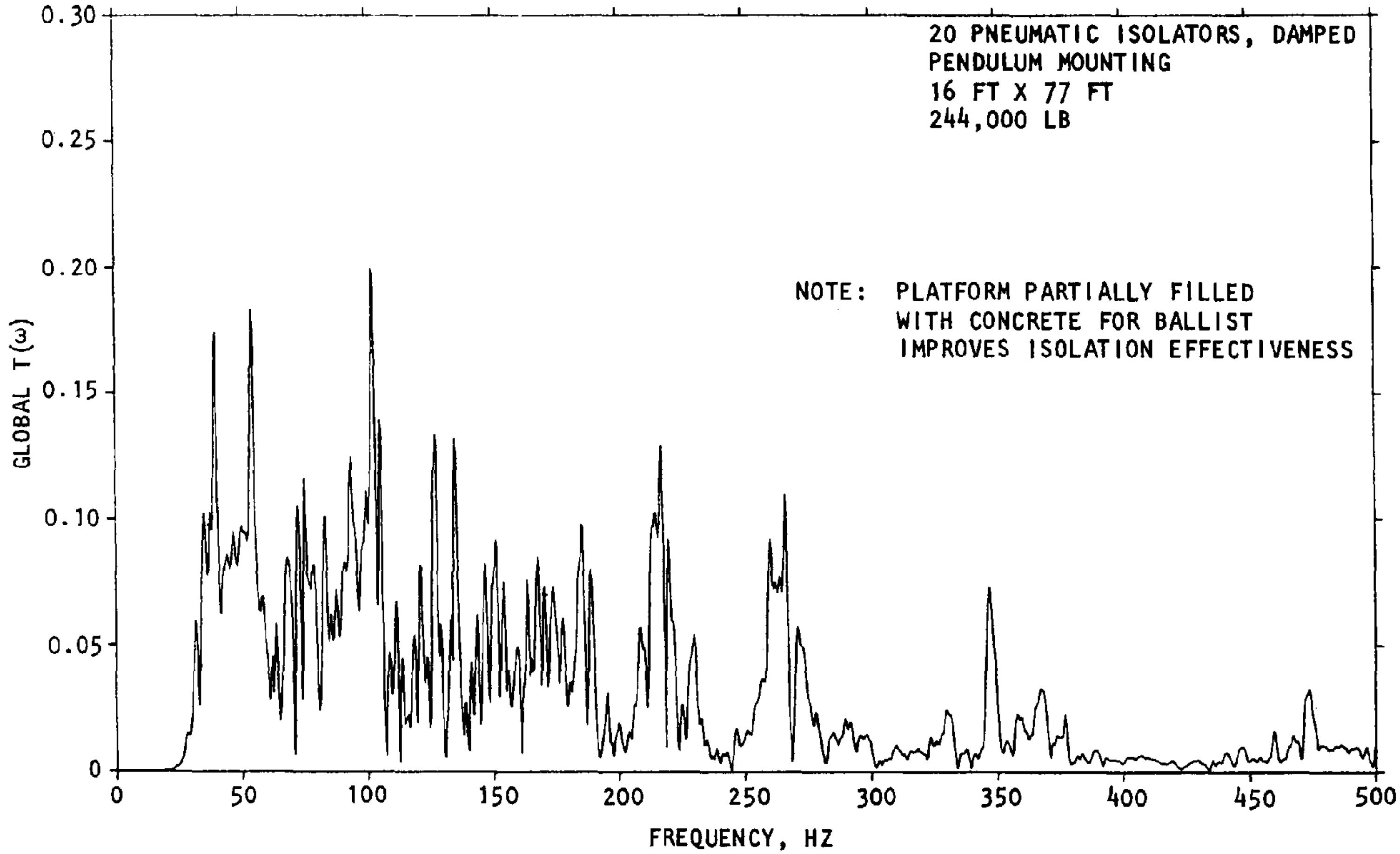


Figure 5-22. Platform MSRPP-2W: Transfer Function Modulus (Safford-Walker, 1975)

CHAPTER 6

SHOCK ISOLATION ELEMENTS

6-1. Introduction.

a. A fundamental element of every shock isolation system is some sort of energy storage or energy dissipative device. These devices must be capable of supporting the mass to be isolated under static and dynamic conditions and, at the same time, prevent transmission of any harmful shock environment to the mass. In most cases the isolation device must have elastic force-displacement characteristics, so that the system will return to a nominal equilibrium position after the support motions cease. Such shock isolation devices or elements are described in this chapter.

b. The desirable features of these devices may be summarized as follows (Saffell, 1971):

—The dynamic force-displacement relationship of the isolator should be predictable for all directions in which stiffness must be provided

—The isolator should have low mass to minimize transmission of high frequency motions to the supported mass

—The frequency of the isolator should remain constant with changes in load, i.e., its stiffness should vary in direct proportion to the load it supports. This allows the system to remain dynamically balanced throughout changes in supported mass and position

—The static position of the isolator element should be adjustable, so that the system can be returned to its nominal position when the suspended load changes

—The isolator element should have high reliability, long life, and low cost

c. The various types of isolation devices used in most protective construction applications possess these characteristics in varying degrees. Any real isolator has some mass; and in some applications, the mass can be quite large. Although this mass is often neglected in preliminary calculations, it must be considered in the final analysis. Nonlinear force-displacement characteristics are often accepted to gain some other advantage.

d. Safford and Walker (1975) suggest that the inclusion of energy dissipative (damping) devices in the isolation system offers several significant advantages, e.g., damping can

—Reduce the severity of output motion response

—Reduce the effect of coupling between modes, thus reducing rattlespace requirements

—Restore the system to an equilibrium position more quickly

—Decrease the sensitivity of the system to variations in input motions

e. Damping can be provided internally in some isolation devices, such as liquid springs, but must be added externally in others, such as helical coil springs. The different types of damping offer advantages and disadvantages that must be considered in the design process. Damping that is proportional to some power of the velocity greater than unity may be effective in attenuating low-frequency components but can increase the severity of high-frequency components. Saffell (1971) states that this characteristic is typical of viscous, hydraulic, and quadratic dampers. Coulomb damping, where the force near the equilibrium position is large, could prevent the system from returning to its nominal equilibrium position. Thus, if degradation of overall isolation system performance is to be avoided, care must be exercised in adding damping devices to a system or in designing isolator elements possessing inherent damping characteristics. Veletsos (1964), Saffell (1971), and Parsons (1962) include detailed discussions of the effects of damping on system response.

f. There are numerous types of devices that can be used to fulfill the shock mitigation function. Commonly used isolators include helical coil springs, pneumatic springs, liquid springs, pendulums, and torsion bars with linear and nonlinear properties. Dampers may be incorporated into the isolating springs or be separate elements, with some providing viscous, Coulomb (friction), or velocity-squared damping. Protective systems have used these individually or in various combinations, including—

—Helical coil springs (horizontal and vertical)

—Pendulum (solid pendant or cable) for horizontal motion with a series-connected helical coil spring for vertical motion

—Pendulum (solid, horizontal) with series-connected pneumatic (vertical)

—Pendulum (solid, horizontal) with series-connected hydraulic (vertical)

g. Isolator elements can be generally classified as passive, active, and semiactive. Platforms, equipment, and weapons can also be considered elements of an entire shock-isolated system.

6-2. Passive isolator elements.

a. Helical coil springs.

(1) The helical coil spring has numerous advantages. It is not strain-rate sensitive, is self-restoring after an applied load has been removed, resists both axial and lateral loads when properly anchored, has a linear spring rate, and requires little or no maintenance.

(2) For most applications, the coil spring requires a large amount of space in comparison to other available shock-isolation systems and the spring cannot be adjusted to compensate for changes in loading conditions. If the weight of a supported object is changed, it may be necessary either to change the spring or add additional springs. For most purposes, the helical coil spring can be considered to have zero damping. If damping is required, it must be provided by external means.

(3) Helical coil springs may be used in either compression or extension. Extension springs are not subject to buckling and may offer a more convenient attachment arrangement. Extension spring attachments, however, are usually more costly and can cause large stress concentrations at the point of attachment. Coil springs are most generally used in compression for shock isolation applications. Buckling, which can be a problem with compression springs, can be prevented by proper design or by the use of guides added either externally or internally to the coils.

(4) Helical coil springs may be mounted in two ways, the ends either clamped or hinged. In most shock isolation applications, the spring ends are clamped, since this method greatly increases the force required to buckle the spring. If space is at a premium, the energy storage capacity may be increased by nesting the springs (placing one or more springs inside the outermost spring). When nesting springs, it is advisable to alternate the direction of the coils to prevent the springs from becoming entangled.

(5) A variety of materials is available for the fabrication of coil springs. The spring can be manufactured from wire as a cold-wound spring or from bars as a hot-wound spring. Hot-wound springs are usually fabricated for wire diameters greater than 0.5 in. (1.27 cm). The specifications for a helical spring may include some or all of the following: maximum load, maximum static deflection, maximum dynamic deflection, spring rate (stiffness), maximum height, maximum diameter, and factors of safety regarding allowable stresses and bottoming of the spring.

(6) The distributed mass of a helical spring has been shown to cause the system to have natural frequencies (spring surge) that are not predicted under the massless spring assumption. For a harmonic excitation whose frequency is near or equal to one of these multiple natural frequencies, the damping present in the system will be the limiting factor governing response. Spring surges selectively pass significant portions of the facility shock input motion into the equipment at the surge frequencies.

(7) The transfer function, $\tau(\Omega)$, for a spring having both viscous and hysteretic damping can be expressed as follows (Mlakar-Walker, 1978):

$$\tau(\Omega) = \frac{(1 + j\eta)(v + ju) + 2\beta j\, q(\Omega)}{(1 + j\eta)(v + ju)\, r(\Omega) - (\Omega + 2\beta j)\, q(\Omega)} \qquad (6\text{-}1)$$

where

η = Shear modulus loss factor of spring material

$u \equiv \sqrt{\dfrac{\sqrt{1+\eta^2}+1}{2\mu(1+\eta^2)}}$

$v \equiv \sqrt{\dfrac{\sqrt{1+\eta^2}-1}{2\mu(1+\eta^2)}}$

$\mu \equiv \dfrac{M}{m}$, ratio of supported mass to mass of spring

M = Mass supported by spring

m = Mass of spring, uniformly distributed over length of spring, ℓ

$\Omega = \omega\sqrt{\dfrac{M}{k}}$ = dimensionless frequency ratio

k = Static spring rate

ω = Angular frequency, driving frequency

$\beta = \dfrac{C}{2\sqrt{kM}}$ = viscous damping ratio

C = Damping coefficient

$q(\Omega) = \sinh v\Omega \cos u\Omega + j \cosh v\Omega \sin u\Omega$

$r(\Omega) = \cosh v\Omega \cos u\Omega + j \sinh v\Omega \sin u\Omega$

Performance characteristics predicted by the above expression for differing amounts of both viscous and hysteretic damping are presented in figure 6-1.

(8) The primary surge frequency and undamped transmission characteristics of a helical spring may be obtained from the following expression (Plunkett, 1958):

$$\begin{bmatrix} F_1 \\ V_1 \end{bmatrix} = \begin{bmatrix} \alpha_{11} & \alpha_{12} \\ \alpha_{21} & \alpha_{22} \end{bmatrix} \begin{bmatrix} F_2 \\ V_2 \end{bmatrix} \qquad (6\text{-}2)$$

where

F_1, V_1 = Input force and velocity of spring

F_2, V_2 = Output force and velocity of spring

$\alpha_{11} = \alpha_{22} = \cos(\omega\sqrt{m/k})$

$\alpha_{12} = j\sqrt{km}\,\sin(\omega\sqrt{m/k})$

$\alpha_{21} = \dfrac{j}{\sqrt{km}} \sin(\omega\sqrt{m/k})$

k = Spring constant

m = Mass of spring

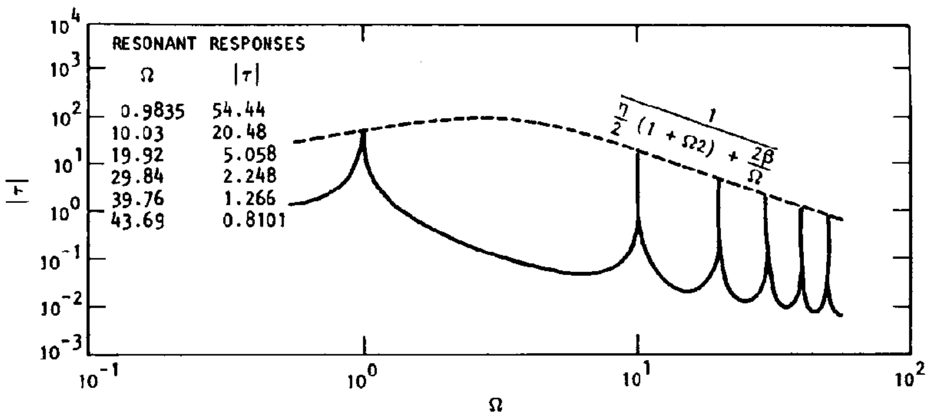
ω = Frequency (rad/sec)

The propagation constant for the spring is:

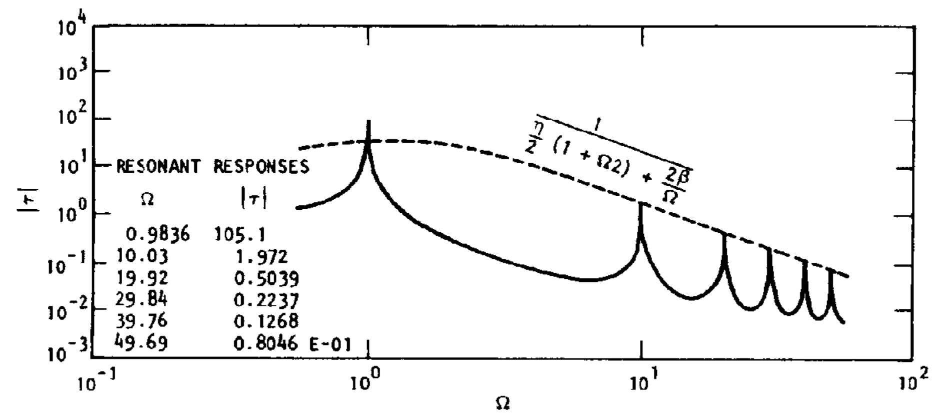
$$\gamma = j\frac{\omega}{c} = \frac{j\omega\sqrt{m/k}}{\ell} \quad (6\text{-}3)$$

where $c = \ell\sqrt{k/m}$, speed of wave propagation of spring of length ℓ

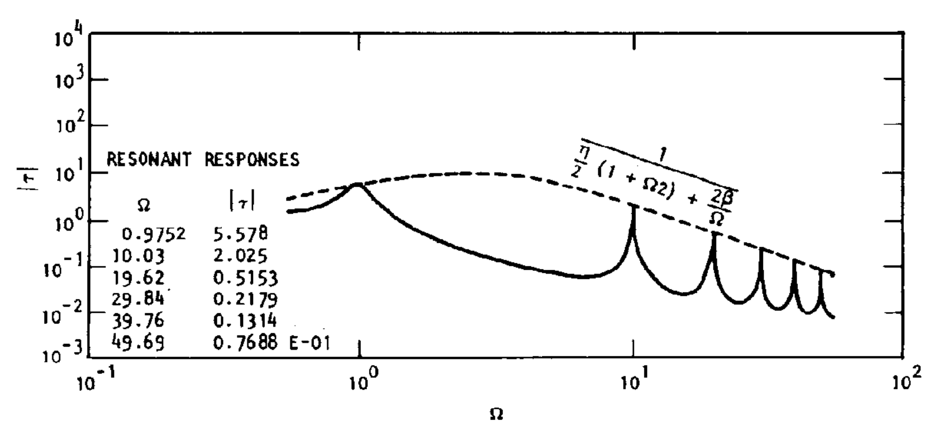
(9) The fixity of the coil spring both to the facility and to the equipment or platform plays a significant



(a) β = 0.01 and η = 0.001



(b) β = 0.01 and η = 0.01



(c) β = 0.1 and η = 0.01

Figure 6-1. Effect of Viscous and Hysteretic Damping on Transfer Function, μ = 10 (Mlakar-Walker, 1978) role, particularly in the high frequencies and where high onset of motion occurs. Poor fixity generates increased transmission of facility motion to equipment at the higher frequencies. An example of this condition is shown in figure 6-2 for isolators of the Minuteman Launch Equipment Room. The poor fixity of the production model results in higher transmissibility, as can be observed in figure 6-3 (Lin, 1977). This poor fixity was the result of "improvements" due to value engineering efforts and were uncovered too late.

(10) There will be serious impacting of a coil spring if there is no end fixity. This situation occurred on the Minuteman Wing I Launch Equipment Room isolator (fig. 6-4). When the onset motion of the facility exceeds the propagation rate of the spring, the spring and its support base separate, leaving the spring to expand and catch up at a subsequent time (Krek, 1970). A series of impacts by the spring are then transmitted to the equipment.

(11) Combined pendulum cables and coil springs have been popular as missile suspension systems. The propagation constant or the limiting velocity of a helical coil spring in expanding from a compressed state leads to problems when coil springs are used with flexible cables. A sudden onset motion of the facility that exceeds the coil spring propagation constant, plus the cable drag loading, creates a slack in the pendulum cable. In the moderate case, a jolt is transmitted to the equipment at a later time when the cable snaps taut. In a more severe case, for example, cable slack permits separation of the isolated platform from the missile it is supporting. A technique used to mitigate this problem is discussed in the section on liquid springs [chap. 6, para. 6-2*c* (12)].

b. Pneumatic springs.

(1) Pneumatic springs are springs whose action is due to the resiliency of compressed air. They are used in a manner similar to coil springs. The two basic types are the pneumatic cylinder with single or compound air chambers and the pneumatic bellows.

(2) Pneumatic springs have the advantage of being adjustable to compensate for load changes. The spring rate can be made approximately linear over one range of deflection and highly nonlinear over another. The springs are quite versatile due to the variety of system characteristics that can be obtained by regulation of air flow between the cylinder chamber and the reservoir tank. These are some of the possible variations:

- —Velocity-sensitive damping by a variable orifice between chamber and reservoir
- —Displacement-sensitive damping by a variable orifice controlled by differential pressure between chamber and reservoir
- —A nearly constant height maintained under slowly changing static load by increasing or decreasing system air content using an external air supply and a displacement-sensitive servo-system controlling inlet and exhaust valves
- —A constant height under widely varying temperature achieved by the same system described above

(3) The disadvantages of pneumatic springs include higher cost and more fragile construction. Reliability of pneumatic springs has been markedly improved by using rolling diaphragms in place of seals. Pneumatic springs have a limited life in comparison to

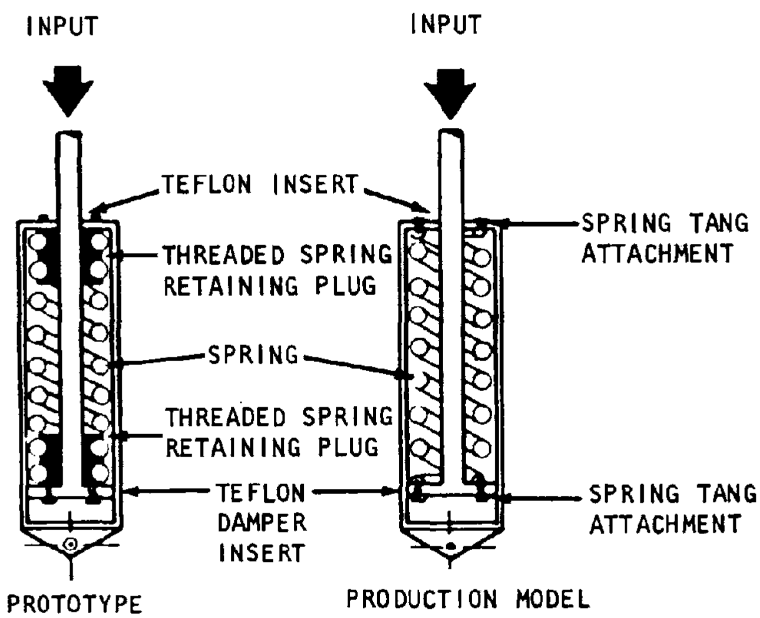


Figure 6-2. Minuteman LER Shock Isolator (Lin, 1977)

mechanical springs and provide resistance only to axial loads. Friction guides transmit motion, and if quadratic damping is used, high-frequency attenuation may be impaired. Orifices in fluid devices normally exhibit damping proportional to the square of the velocity. This effect limits excessive excursions at the major resonances of the "rigid" platform isolation system. At the higher frequencies, this type of velocity-squared damping will transmit more facility motions into the platform. However, the magnitudes of the higher frequencies of airblast loads (fig. 3-3) and ground shock are considerably lower; thus quadratic damping effects at the higher frequencies become much less severe, and these devices may often be used very effectively.

(4) Pneumatic cylinders are fabricated in the two basic configurations illustrated in figure 6-5. The spring rate of the single action cylinder is nonlinear, with an increasing spring rate for positive deflections of the piston and a decreasing spring rate for negative displacements. The double-action cylinder also exhibits a nonlinear spring rate with an increasing rate for both positive and negative displacements. These characteristics are illustrated graphically in figure 6-6, which is a plot of the stiffness ratio versus the displacement ratio. The displacement ratio is defined by

$$\eta = y/L_o \qquad (6\text{-}4)$$

where

y = Deflection

L_o = Cylinder chamber length (see Fig. 6-5, $L_{o1} = L_{o2} = L_o$ for a double-action cylinder at the neutral position)

Note that the spring rates are approximately linear for single-action cylinders and almost constant for double action cylinders at displacement ratios less than 0.3.

(5) Under large displacements, spring rates become nonlinear but at a gradual rate (Fox-Steiner, 1972). This nonlinear effect can be beneficial for equipment survival, where there are uncertainties of input displacements of the facility.

c. *Liquid springs.*

(1) A liquid spring consists of a cylinder, a piston rod, and a high-pressure seal around the piston rod. The cylinder is completely filled with a liquid; as the piston is pushed into the cylinder, it compresses the liquid to very high pressures.

(2) The configurations of liquid springs are divided into three major classes according to method of loading. The classes are simple compression, simple tension, and compound compression tension. Although they are loaded in different ways, all three types function by compressing the liquid in the cylinders. Schematics of the tension and compression types are shown in figure 6-7 and in combination as shown in figure

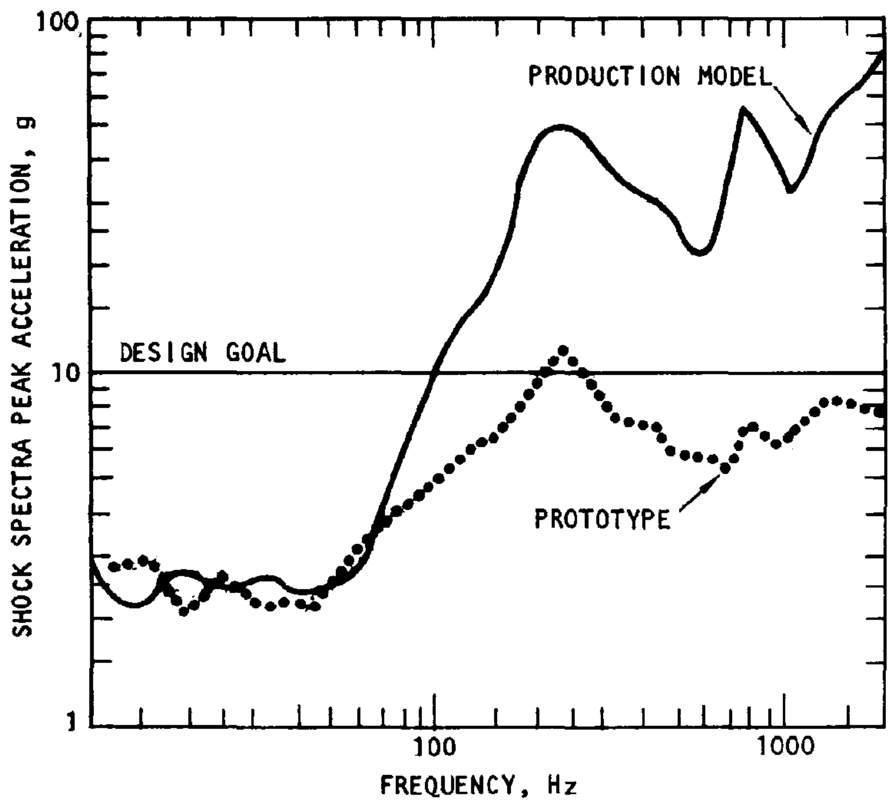


Figure 6-3. Isolator Transmissibility (Lin, 1977)

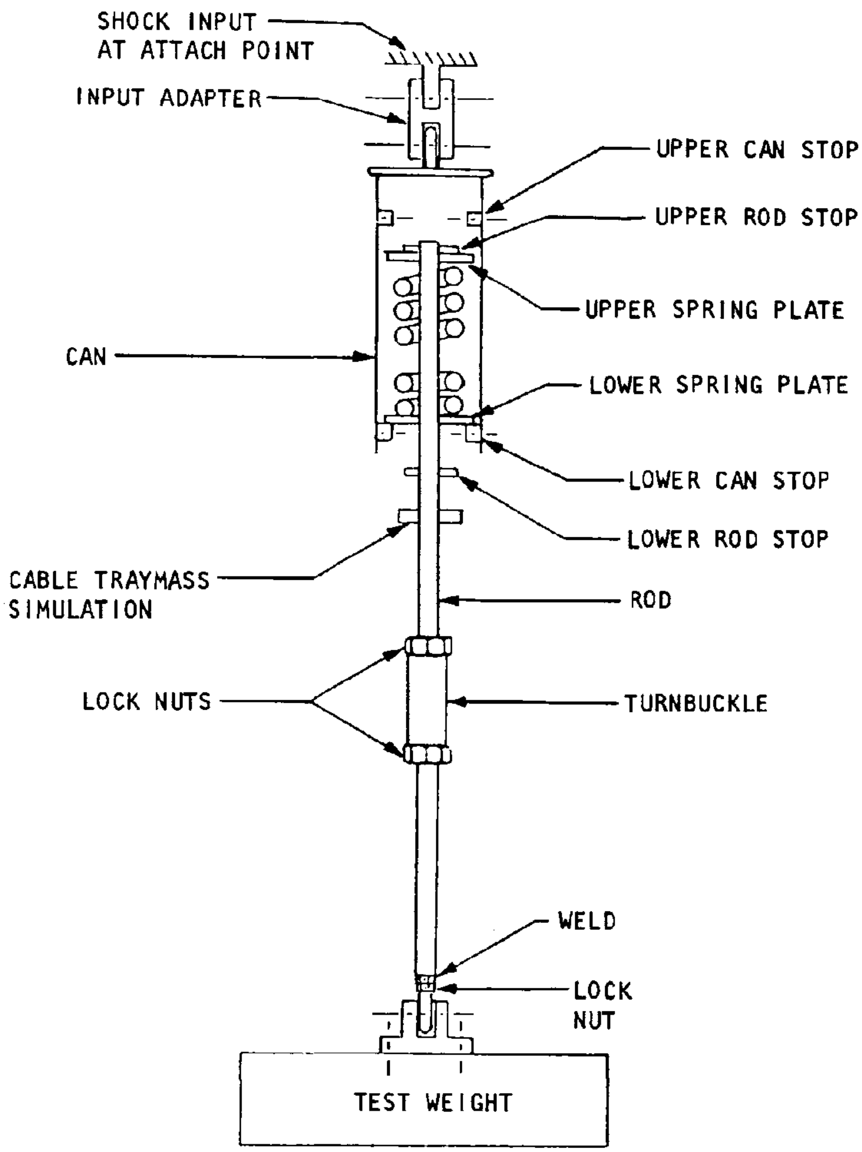


Figure 6-4. Minuteman Wing I LER Isolator Assembly in Test Configuration (Krek, 1970)

6-10. The tension type is more common in protective construction applications. The compound spring is merely a more complex mechanical combination of the two basic types. The cylinders are often fitted with ported heads to guide the pistons and provide damping. Damping can also be provided through by adding drag plates to the piston rods.

(3) Liquid springs are very compact devices with high, nearly linear, spring rates. They can be adjusted to compensate for load changes, are self-restoring, and can absorb large amounts of energy. They are highly sensitive to changes in temperature and in fluid volume. The latter usually results from fluid chamber leakage or expansion under pressure. Because liquid springs normally operate at high pressures, high-quality, close-tolerance seals are required around the piston. Friction between the seal and piston provides appreciable damping and increases the spring rate from 2 to 5 percent. Liquid springs are high-pressure vessels requiring high quality materials and precision machine work; as a result, they are normally expensive. However, they are difficult to equal as compact energy-absorption devices.

(4) The high operating pressures in liquid springs require close tolerance seals around the piston rod, as noted above. Effective sealing is accomplished by pressure between the seal and the piston rod. These pressures in turn produce a friction deadband that opposes isolator motion. The friction deadband is equal to twice the friction force acting on the rod and must be overcome to initiate isolator rod motion. Input forces less than this friction force will be transmitted with-

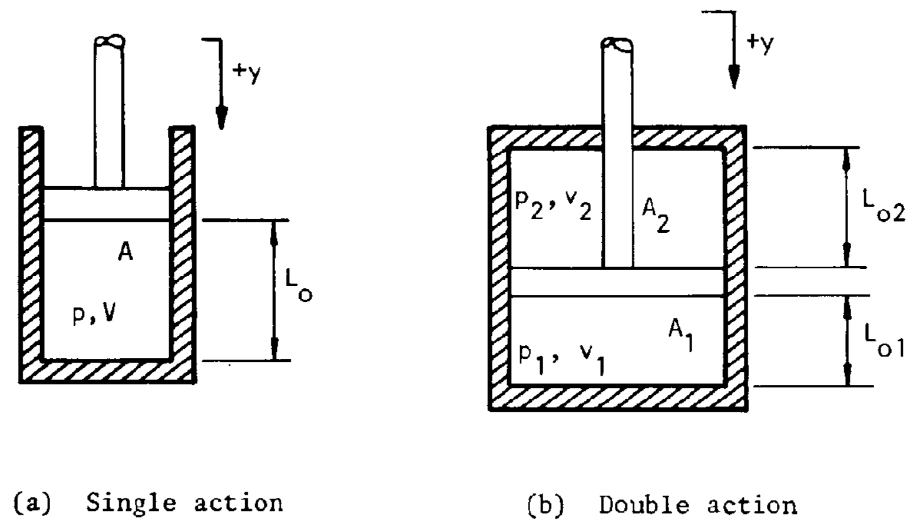


Figure 6-5. Schematic of Pneumatic Piston Springs. (Crawford et al., 1974)

out significant attenuation to the isolated mass. The friction deadband can also affect the final position of an isolated item after its response to input motions. In multiple isolator systems, the stopping of isolators at different positions could result in undesirable tilt or displacement of the system. In general, the effect of a given friction deadband on final system position decreases with increasing isolator spring rate. An increase in isolator internal pressure will normally increase the width of the friction deadband.

(5) Since the liquid spring is strongly dependent on the characteristics of the fluid in the cylinder, it is important that the fluid selected has the best combination of desirable properties. The desired properties include:

—Suitable viscosity
—Temperature and chemical stability
—High compressibility

The viscosity of the fluid is important for two reasons: First, if the viscosity is too low, it may be difficult to maintain a good pressure seal around the piston rod; and second, the damping characteristics of the spring are directly related to the fluid viscosity.

(6) If the spring is to be subjected to temperature

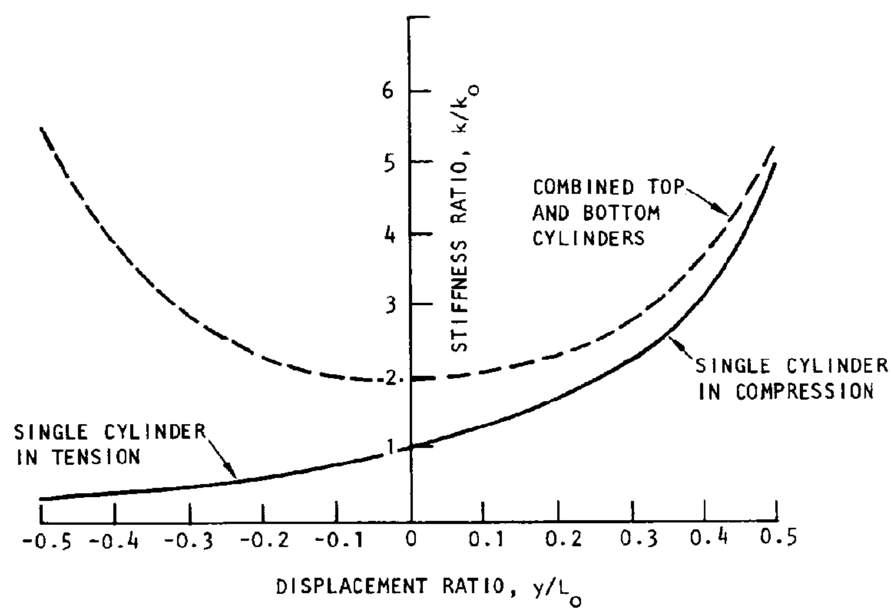


Figure 6-6. Stiffness Variation with Displacement (AJA, 1966)

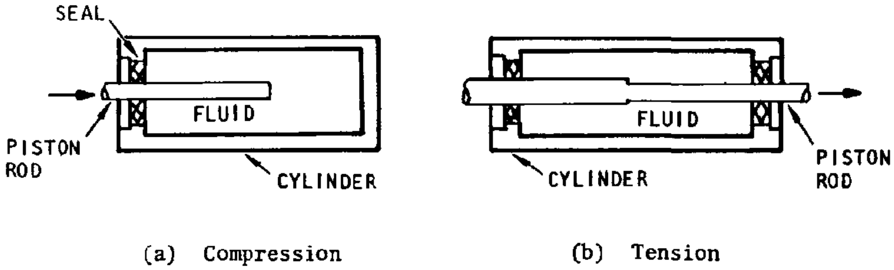


Figure 6-7. Liquid Spring Schematic (AJA, 1966)

changes, the coefficient of thermal expansion for the fluid is important, since fluid volume changes will result in pressure changes. The coefficients of thermal expansion for most fluids are decreased by the operating pressures normally encountered in liquid springs. Temperature changes will not be a problem in most shock isolation applications. If necessary, steps can be taken to control the fluid temperature as, for example, with the use of thermal blankets.

(7) The compressibility of the spring fluid directly affects the volume of fluid required for a specific application. The higher the compressibility, the less fluid required. The compressibilities of several typical liquid spring fluids are shown in figure 6-8. The silicone oils, which include the three lower curves, provide the best combination of desirable properties and are the usual selections for liquid springs. Although Dow Corning's 210 with a viscosity of 0.65 centistoke has the highest compressibility, its low viscosity could cause sealing problems. Military specifications are available that describe fluids suitable for use in these devices.

(8) The design criteria for liquid springs normally include the maximum spring rate, the static displacement, and the peak dynamic displacement. For a preliminary design, the frictional forces, temperature effects, and volume changes in the chamber due to fluid pressure are generally neglected. A maximum pressure and preload pressure should be selected based on the spring rate and peak dynamic displacement.

(9) Liquid springs have been successfully used for isolation systems in the Minuteman system. Additional liquid spring development occurred with the Minuteman upgrade system. Liquid spring configurations studied for Minuteman are shown in figure 6-9, and the final configuration is shown in figure 6-10 (Ashley, 1976).

(10) Performance of these liquid springs under

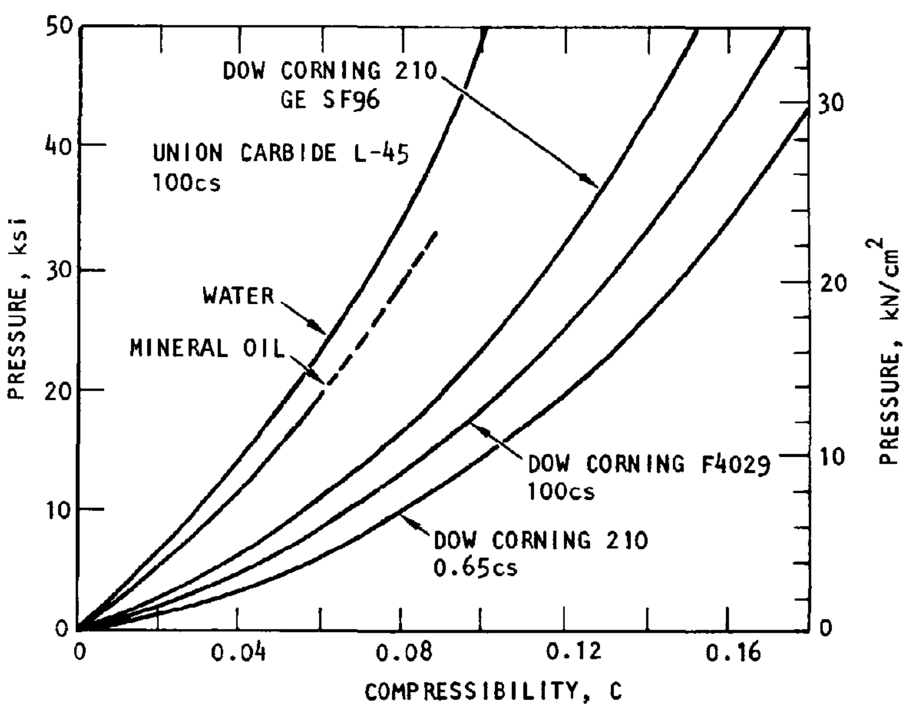


Figure 6-8. Compressibility of Fluids (AJA, 1966)

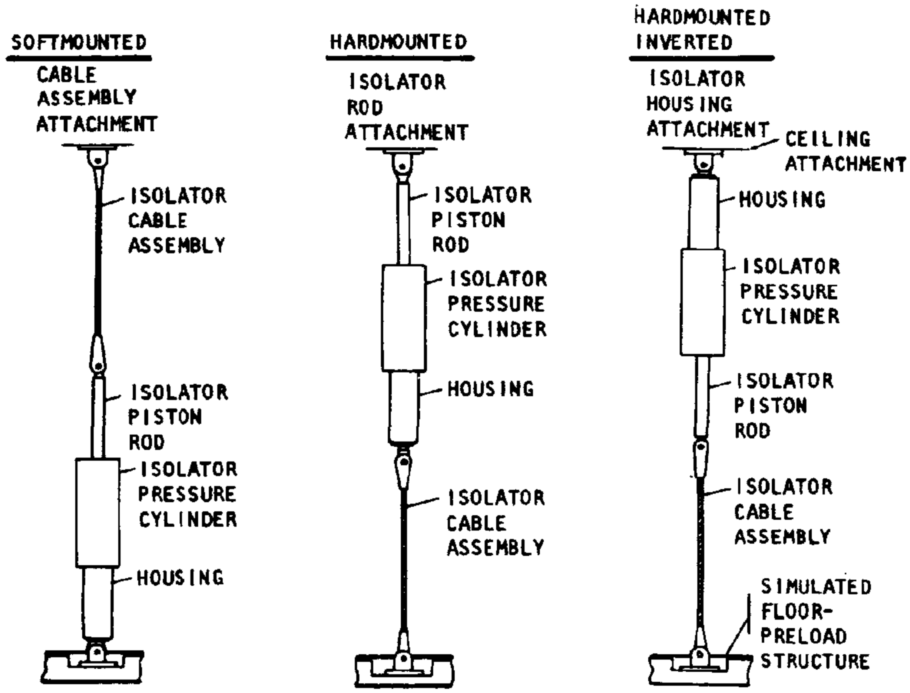


Figure 6-9. Basic Structural Arrangements, Liquid Isolator Development Tests (Ashley, 1976)

high impulse loads while supporting a dead-weight load is shown in figures 6-11 to 6-14. The transmissibility for these plots is defined as a shock spectrum (Q = 5) ratio and not as a standard ratio of Fourier spectra. Structural arrangement, damper valve, and preload are major contributors to the performance characteristics shown in the figures.

(11) Seal friction also affects transmissibility; for the candidate isolator this amounts to ±500 lb. The significance of seal friction with respect to the preload can be seen in figure 6-13 and in table 6-1.

(12) An interesting application of a liquid spring in series with a helical coil spring connected to a pendulum mechanical cable was made for shock isolation of an ICBM. The configuration of this device is shown in figure 6-15; performance is given by figure 6-16. In this application, the liquid spring was bottomed out in order to take up or minimize cable slack in the pendu-

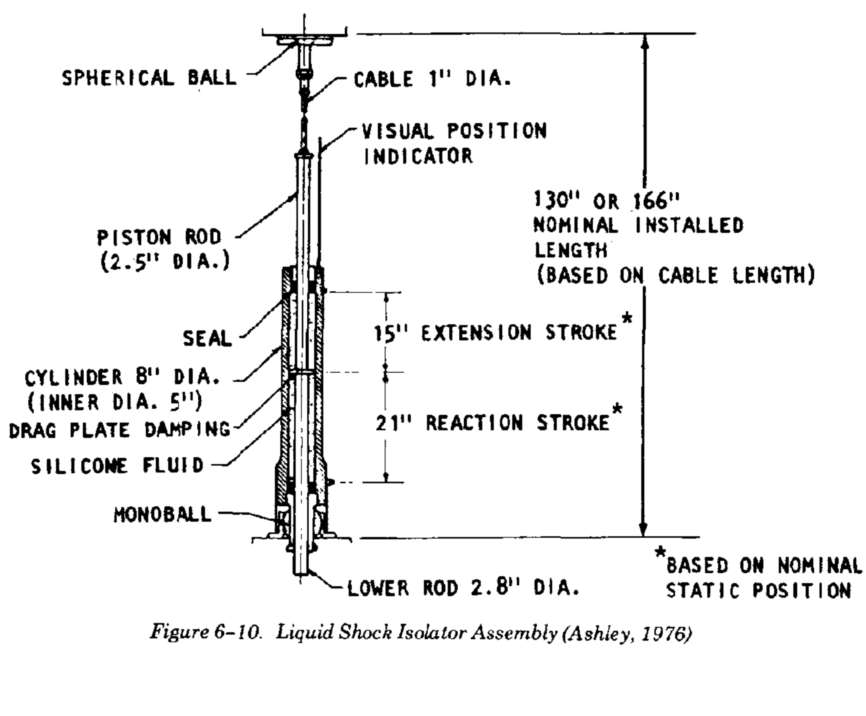


Figure 6-10. Liquid Shock Isolator Assembly (Ashley, 1976)

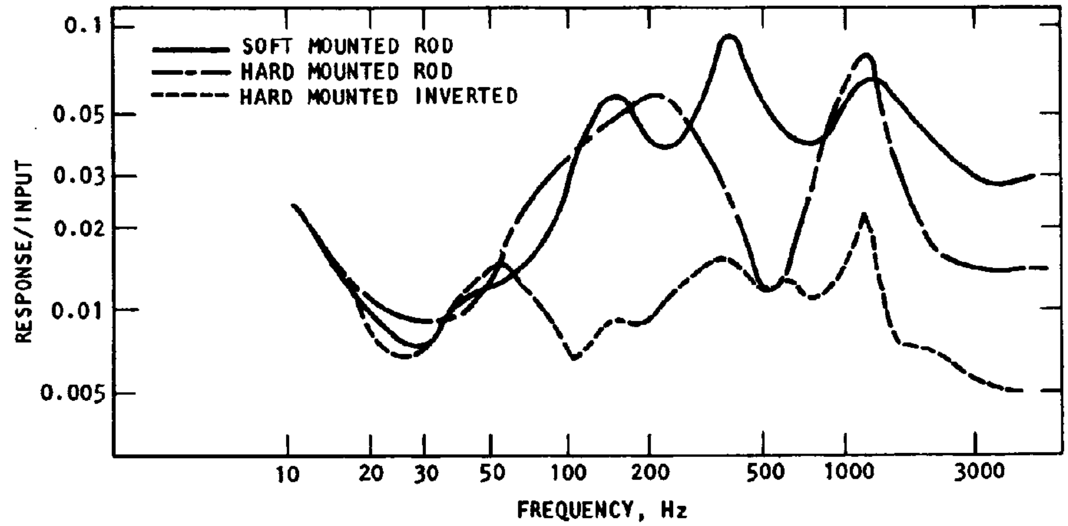


Figure 6-11. Q = 5 Shock Spectra Ratios, Candidate Isolator Transmissibility Characteristics (Ashley, 1976)

lum in case of sudden downward motion.

(13) Following tests with the figure 6-15 configuration, tests were conducted on a prototype missile isolation system (fig. 6-17). The transfer function for this test prototype is shown in figure 6-18. Note that as more structure and elements are included in the system, there is a loss in isolation efficiency and the curves of figure 6-18 begin to look like the platform curves of figures 5-13 to 5-22 in chapter 5. More detailed information may be obtained from Mazur's *Liquid Spring Design* Data (1970).

(a) Liquid (or hydraulic) springs should not be viewed exclusively as energy storage devices with force proportional to displacement. These springs include fluidic effects together with nonlinear spring and damping effects that permit a wide range of performance characteristics (fig. 6-19). As discussed for pneumatic isolators, hydraulic springs are available with controlled hardening of spring rates at the limits of stroke to prevent impact bottoming. In fact, considerable latitude is permitted in specifying force-displacement curves.

(b) Performance of liquid springs and other high-performance isolators can be defined in terms of dissipated energy—that is, by force-displacement relationships. For example, equipment fragility imposes limits on acceleration, thus controlling the maximum force transmitted to the equipment through the isolator. Either overall or internal displacements can impair functional survival, so both types of displacement must be constrained. The dotted horizontal and vertical lines in figure 6-20 indicate the performance of

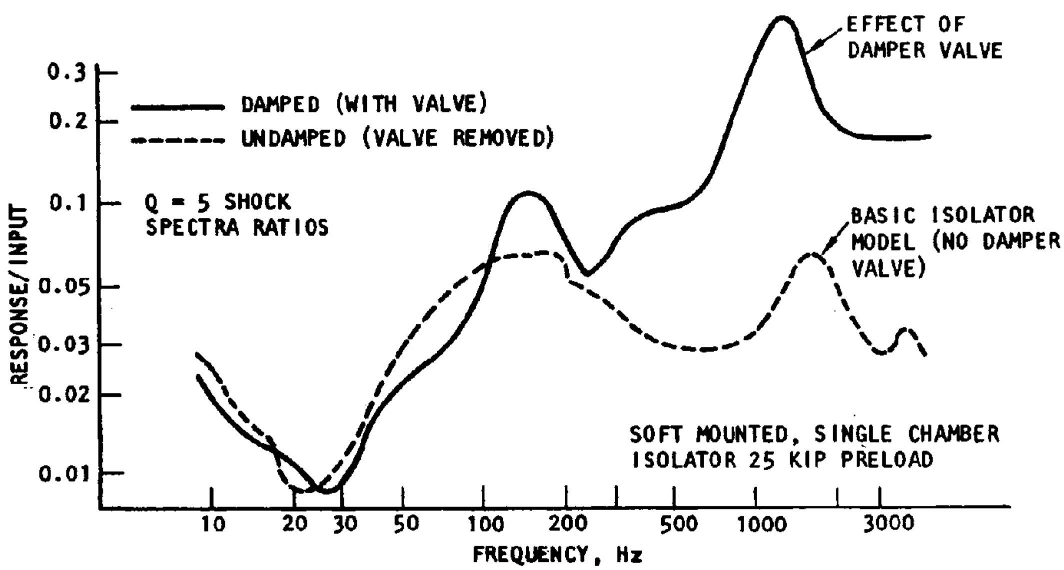


Figure 6-12. Effects of Damping Valve, Candidate Isolator Transmissibility Characteristics (Ashley, 1976)

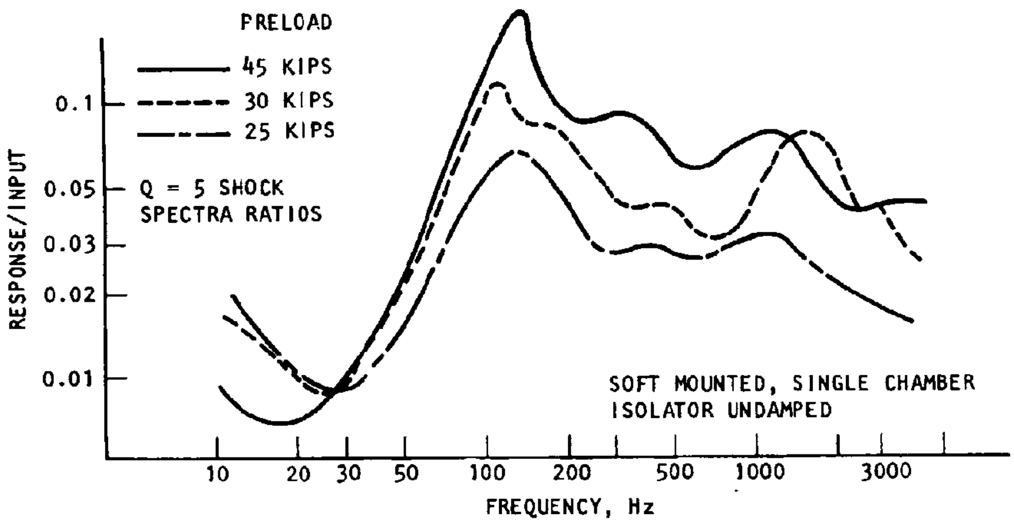


Figure 6-13. Effects of Isolator Preload, Candidate Isolator Transmissibility Characteristics (Ashley, 1976)

an ideal isolator: one that dissipates as much energy as possible. The solid line is typical of actual isolator performance.

(c) The development of liquid or hydraulic isolators having the above characteristics has been largely due to the application of fluidics in damping control. The elements that enter this design process are illustrated in figure 6-21 (from Taylor, 1971). Residual stored energy in the system can be specified to control the rate of return to equilibrium of the isolated platform or equipment. Some offset and tilt will also occur due to seal hysteresis.

d. Torsion springs

(1) Torsion springs provide resistance to torque applied to the spring. In shock isolation applications, the torque is usually the result of a load applied to a torsion lever that is part of the torsion spring system. A typical torsion spring shock-isolation system is illustrated in figure 6-22.

(2) Since the axis of a torsion spring is normal to

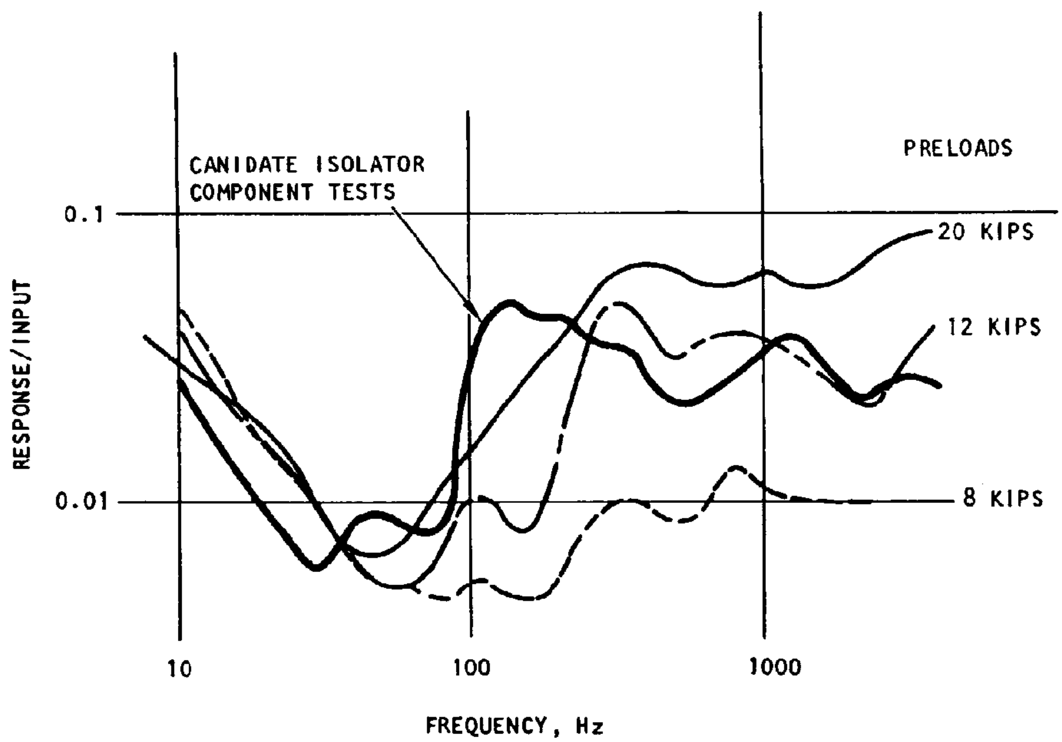


Figure 6-14. Shock Spectra Ratio for High Level Input Positive Pulse (Ashley, 1976)

TABLE 6-1. LIQUID SPRING CHARACTERISTICS
(Ashley, 1976)

Preload Configuration, kips	Liquid Spring Rate, lb/in.	Rigid-Body Frequency, Hz
8	600	0.84
12	650	0.74
20	810	0.65

Seal Friction = ±500 lb
Dragplate Damping (C_V) = 0.15

the direction of displacement, it can be used advantageously when the space in the direction of displacement is limited. Torsion springs have linear spring rates, are non-strain-rate sensitive, are self-restoring, and require little or no maintenance.

(3) Torsion springs cannot be adjusted to compensate for changes in weight of shock-isolated equipment. Also, damping must be provided by external means. The axial length of some types may preclude their use when space is limited.

(4) There are three basic types of torsion springs: torsion bars, helical torsion springs, and flat torsion springs. The type to use will depend on the space available and the load capacity required. The torsion bar is normally used for light to heavy loads, the helical torsion spring for light to moderate loads, and the flat torsion spring for light loads. The torsion bar is the type most commonly found in protective structure applications. Crawford et al. (1973) describe procedures for design.

(5) The torsion bar is most commonly used where large loads must be supported. A typical bar is illus-

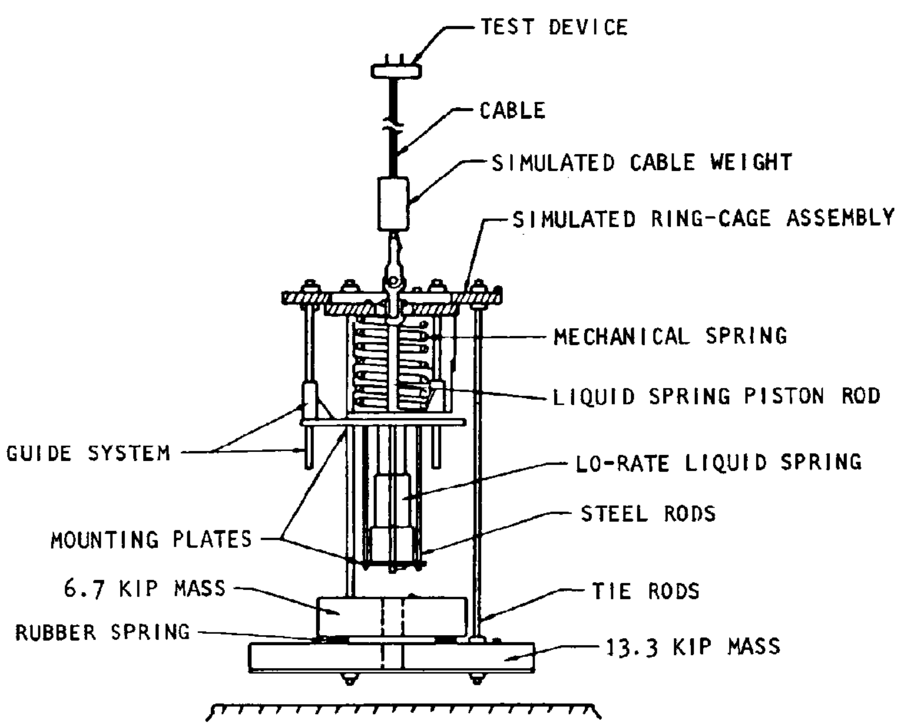


Figure 6-15. Test Setup for Concept Development Shock Testing of a Liquid/Mechanical Isolator (Ashley, 1976)

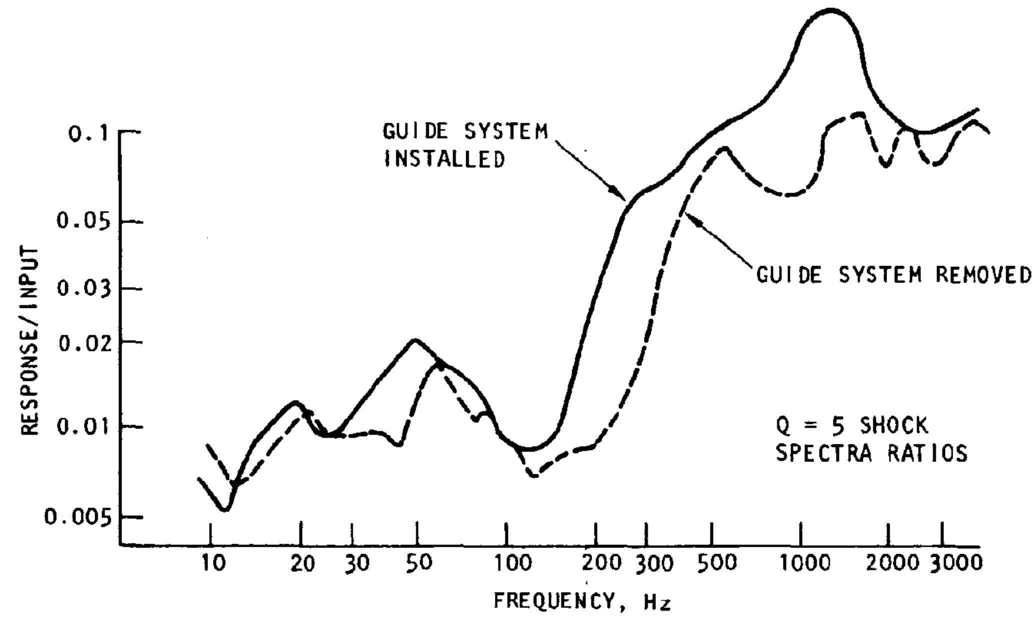


Figure 6-16. Vertical Transmissibility Liquid/Mechanical Isolator Test Results (Ashley, 1976)

trated in figure 6-23. On the basis of the ratio of energy storage to device weight or volume, the torsion bar is one of the most efficient spring systems. Its diameter and length can be varied to suit the applied load and available space. For most practical purposes, it may be considered to have zero damping, although damping may be added to the shock-isolation system through the support strut assembly.

(6) To minimize bending stresses and increase service life, torsion bars should be supported at both ends and fitted with bearings at the end where rotation takes place. This entails the additional expense of bearing-support and bar-anchoring systems. The ends of torsion bars are usually splined to facilitate connections; large stress concentrations can occur in the vicinity of these splines.

(7) The maximum applied load and allowable deflection must be specified before a torsion bar can be designed. The maximum allowable stress in the bar will depend on the material used for the bar, the fabrication process, and the anticipated loading conditions.

e. Other passive isolator elements. The helical, torsion, pneumatic, and liquid springs are the more common types of isolators for large masses. There are many other devices especially suited for particular applications and smaller loads. The following paragraphs briefly describe some of these isolators, with references to additional sources of information.

(1) Plastic foams.

(a) Although not shock isolation devices, plastic foams have been extensively used in the packaging of shock-sensitive items and have also been considered for use in some protective construction applications. They are classified as flexible or rigid foams and open or closed cell. The flexible foams exhibit much smaller permanent deformations under transient loads than rigid foams of the same strength. Both types show an increase in strength with increasing density. Their stress/strain characteristics are highly nonlinear and both types can undergo large (40 to 60 percent) strains before the onset of strain hardening. Figure 6-24 shows the load/strain relationships for a typical polyurethane foam. The foam of figure 6-24 indicates both energy storage and energy dissipative behavior. The stress/strain properties of some plastic foams are also sensitive to rate of loading and temperature.

(b) The energy-storage and energy-dissipative behavior in open-cell plastic foams comes from the viscoelastic properties of the material and the presence of entrapped air. The air has a marked effect on the mechanical behavior because of the airflow resistance and pressure buildup under dynamic conditions. As the foam is compressed, the cell pores are reduced in size, causing a change in flow resistance. The restricted flow causes hardening of the foam due to air pressure buildup, especially at high strain rates.

(c) Foam isolators offer considerable design flexibility. They can be formed into almost any geometric shape and, hence, configured to avoid local hard points and attendant design problems. Tests on isolator elements subjected to typical nuclear-weapon-induced facility shock pulses showed no discernible anomalous behavior nor surging, and high frequencies were effectively attenuated (Liber et al., 1969). Flexible foams can be used for multiple-shot applications, although there is some degradation with cyclic loading. Liber et al. (1969), Voltz (1967), and Liber-Epstein (1969) report on investigations of the use of flexible

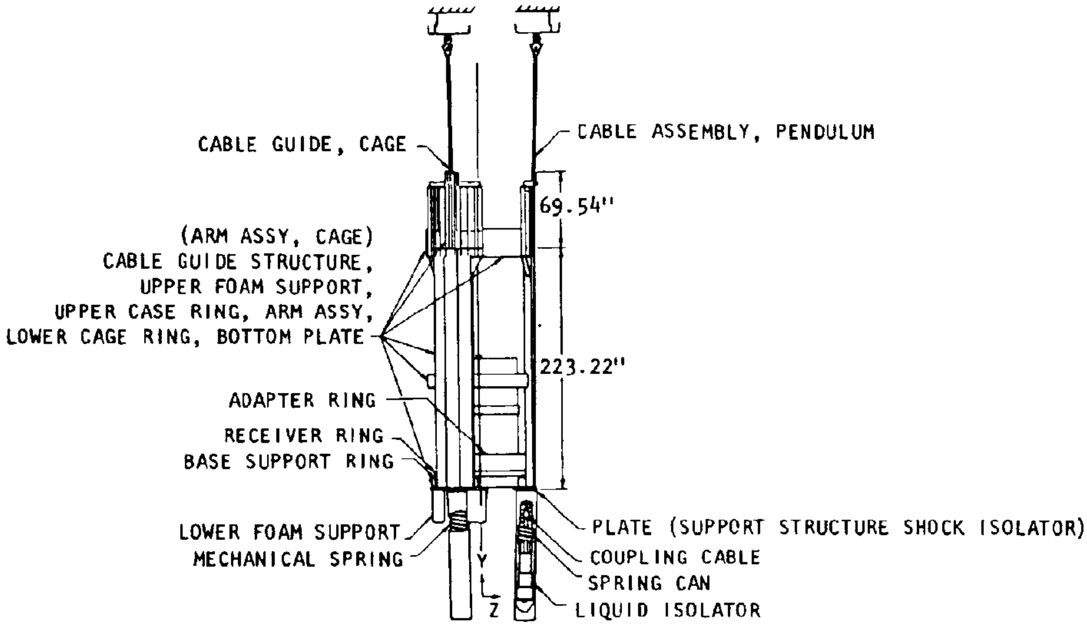


FIGURE 6-17. TEST SETUP FOR MISSILE SUSPENSION SYSTEM SINGLE ISOLATOR DEVELOPMENT TEST PROGRAM (Ashley, 1976)

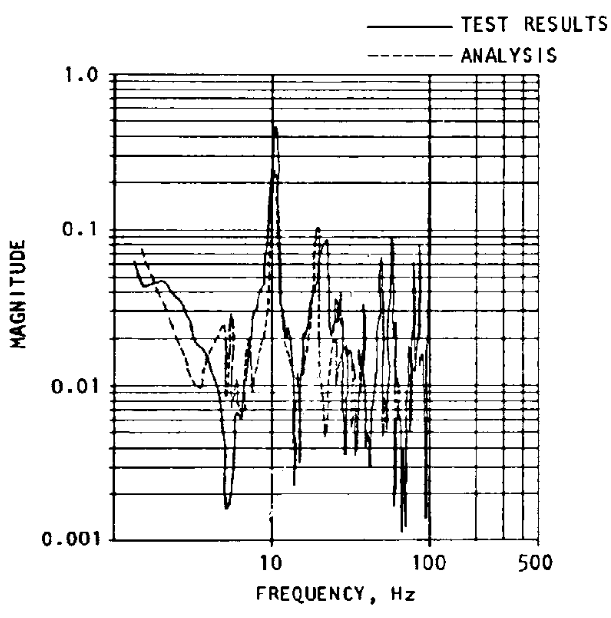
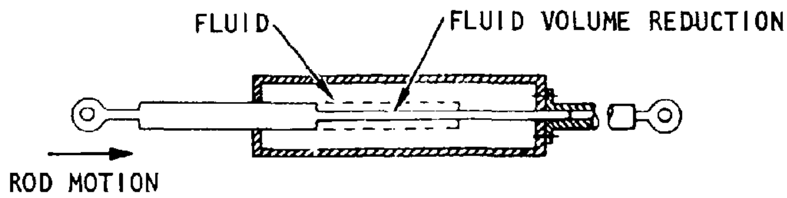
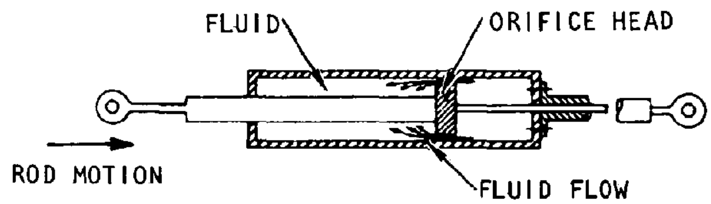


FIGURE 6-18. TEST-ANALYSIS COMPARISON OF FOURIER TRANSFER FUNCTIONS BASE SUPPORT RING TO INPUT (Ashley, 1976)



(a) Spring effect



(b) Damping effect

FIGURE 6-19. CHARACTERISTICS OF A LIQUID-SPRING SHOCK ISOLATOR (Sonnenburg et al., 1977)

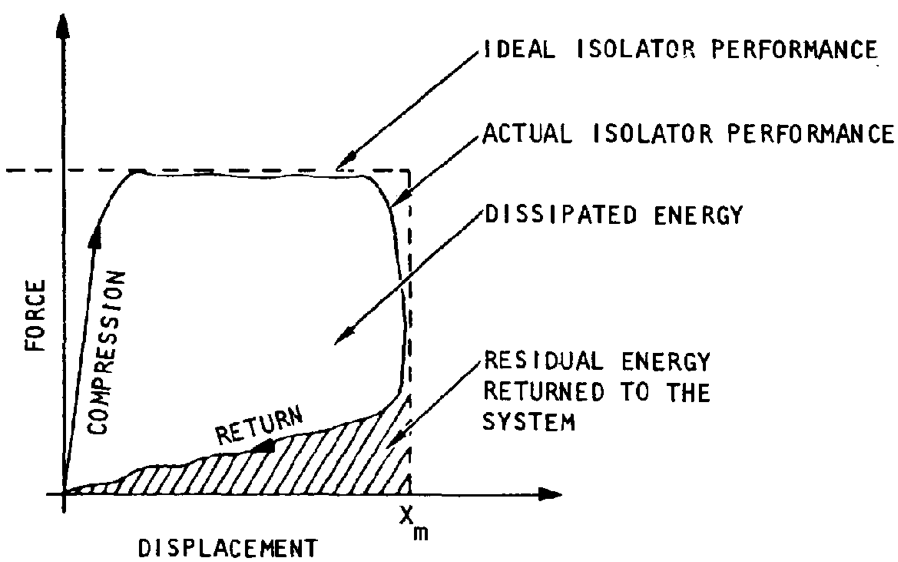


FIGURE 6-20. FORCE-DISPLACEMENT RELATIONSHIP FOR HIGH-PERFORMANCE ISOLATORS (Sonnenburg et al., 1977)

open-cell polyurethane foams for shock isolation of hardened underground facilities.

(d) The distributed nature of foam isolators is attractive for uniform symmetric geometries and mass properties. Such isolators meet the requirement of omnidirectional force-displacement behavior with minimum translational-rotational coupling. However, for nonsymmetric geometries and mass properties, a design problem arises in maintaining the force resultant close to the article c.g. to minimize coupling. For this case, it may be necessary to use a set of properly configured discrete foam isolators or some other technique to tailor the distribution of isolator forces.

(e) For an isolation system consisting entirely of foam, another potential problem is the long-term creep due to the steady gravity load. If creep is a problem, foams may still be attractive for lateral isolation in conjunction with other isolators to handle vertical loads.

(f) Polyurethane isolation was used in an early application on the SNARK missile. The purpose of this application was to protect delicate elements such as frequency standards and tape drives during rocket launch and cruise (Safford-Inouye, 1957). Additional applications to avionics are covered in the Arthur-Carrell report (1969).

(g) Hardness upgrade of an ICBM system required addition of plastic foam in the rattlespace (fig. 6-25) of a shock-isolated platform and around the shock-isolated missile mount (fig. 6-17). These studies at the Boeing Aircraft Co. provided further dynamic performance information on foams that corroborated work described in Liber et al, (1969). This work covered the effects of preworking the foam (fig. 6-26), of preload (figs. 6-27, 6-28), and of loading rates (fig. 6-29).

(h) Referring to figure 6-26: Open-celled polyurethane foam is made up of cells and connecting

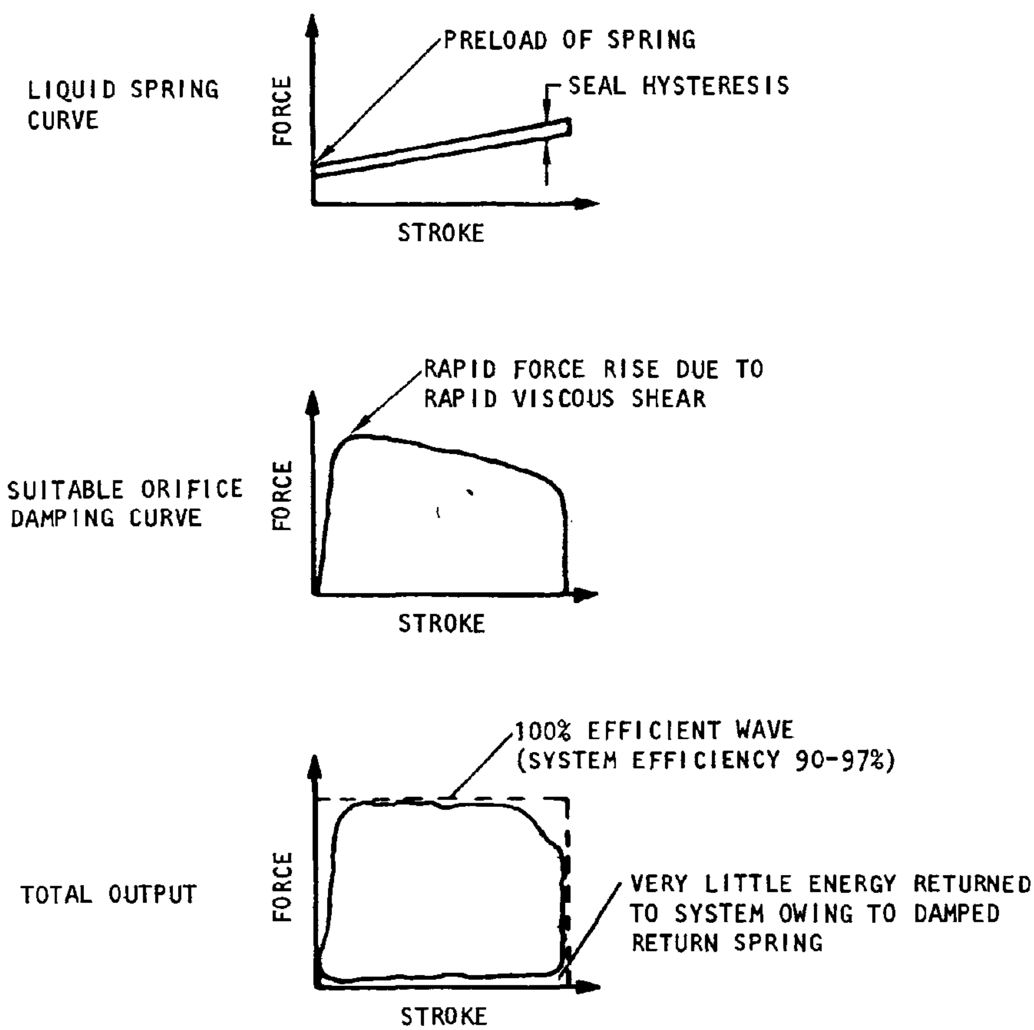


Figure 6-21. Performance Characteristics of Constant Force Liquid Isolator (Taylor, 1971)

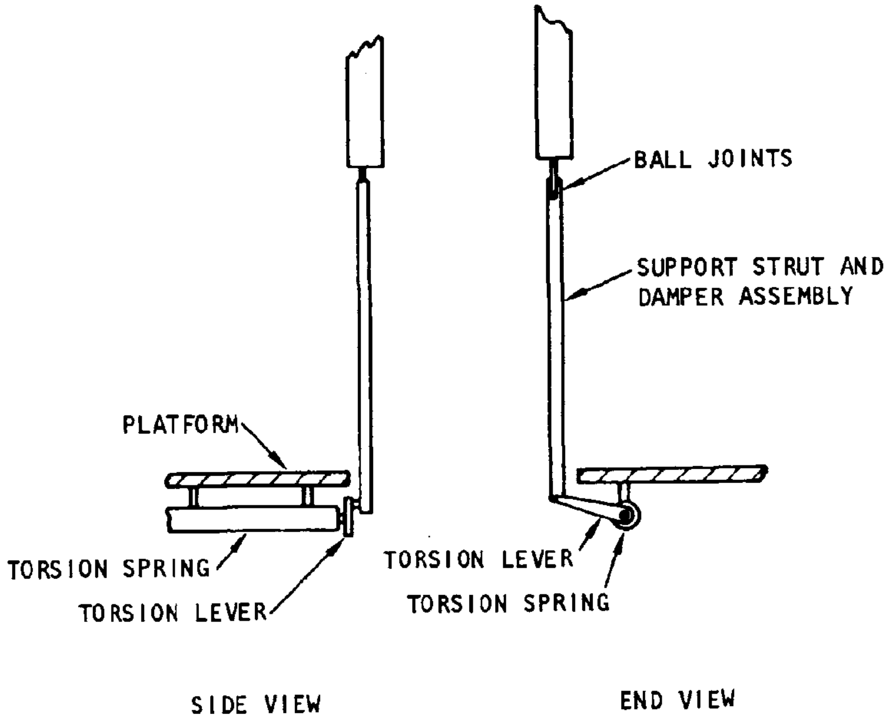


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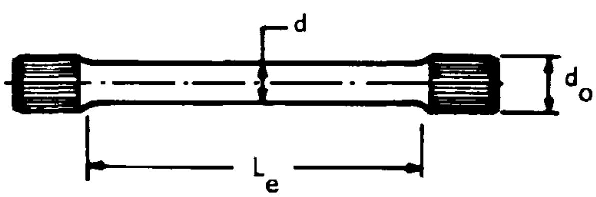
Figure 6-22. Typical Torsion-Spring Shock Isolation System



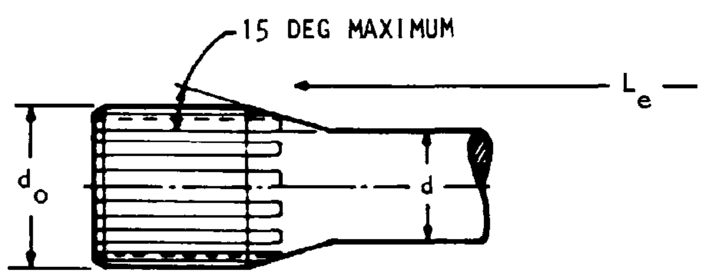


Figure 6-23. Standard Single Torsion Bar with Splined Ends (AJA, 1966)

structure. The density of the foam is a function of cell-wall thickness and cell size. When foam is statically crushed, there is a certain cellular wall breakdown to the extent of the amount of deflection. Figure 6-26 gives a good example of this phenomenon. Once the foam has been precrushed, the force deflection characteristics remain fairly constant for a limited number of cycles, if the foam is allowed time to regain the air that was pushed out of it. Repeated cycles during a short period of time show a reduction in force for a given deflection. This force reduction is due to the loss of air that has been pushed out and has not had time to return.

(i) The significance of the Boeing investigations is in the size of the equipment protected, the magnitude of the weapon-effect threats, the test verifications of the foam isolation systems, and the application to an operating weapons system. Application studies for the ICBM are reported in Ashley (1976), Gustafson (1976), Milne (1976), Mortimer (1977), and Luschei (1977).

(j) An additional class of device that may be considered for installations similar to the above is the marine dock bumper. "Portslide," a type manufactured by the General Tire and Rubber Company, consists of a butyl rubber body covered by a cast low-friction urethane surface. Another type is the air-block rubber fender manufactured by the Yokohama Rubber Company, Mitsubishi International Corporation.

(2) Disc springs.

(a) Disc springs are washer-shaped discs of spring temper metal. They have a strong spring action that resists axial deformation when loaded axially, in opposite directions, at the inner and outer edges. The

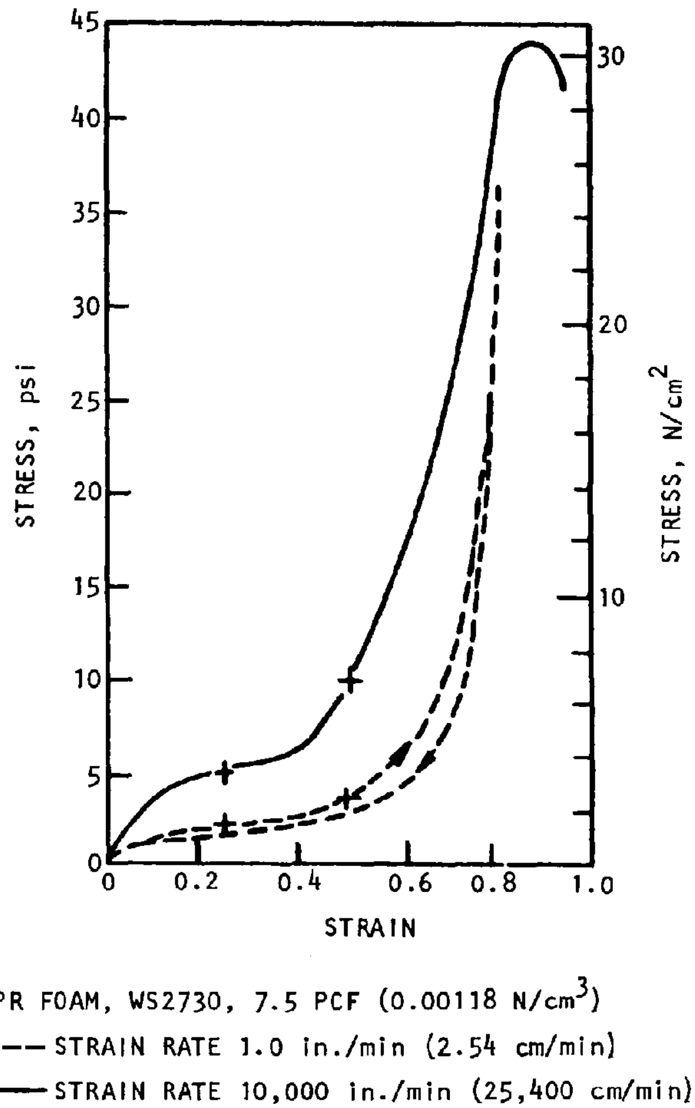


Figure 6-24. Stress/Strain Curve for CPR, WS2730, 7.5 PCF Polyurethane Foam (Liber et al., 1969)

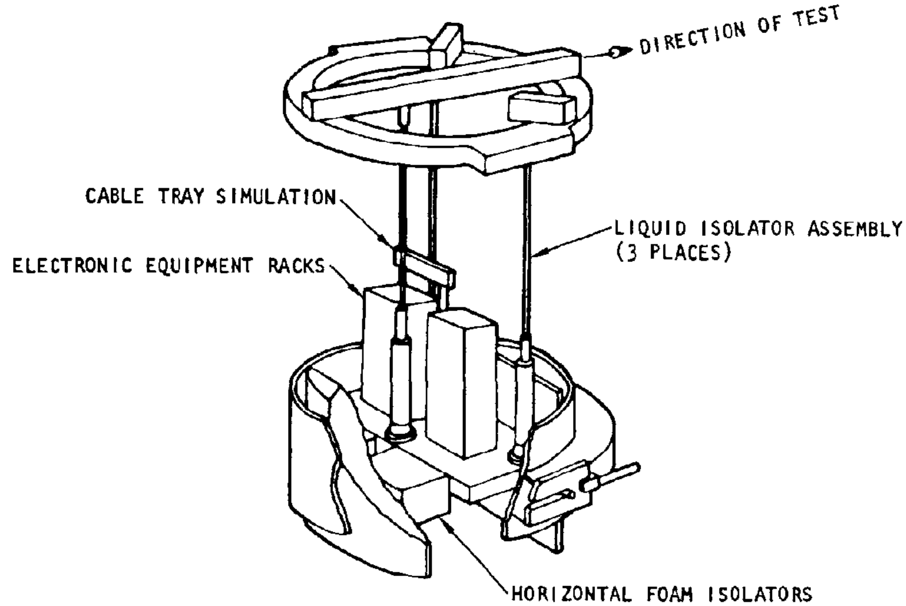


Figure 6-25. Floor Segment Test Specimen Schematic (Gustafson, 1976)

basic and most widely used disc spring element has a uniform metal thickness. A useful variation is radially tapered in thickness so that, as a full disc, the thickness at the center would be zero (fig. 6-30). Radially tapered disc springs are more efficient in terms of energy-storage capacity per unit weight (or volume) of metal, but are not widely used due to their higher manufacturing costs. Both forms are of interest because of their inherently high efficiency and their very high feasible total load capacity as compared to helical coil springs.

(b) The main advantage of disc springs over other types of springs is their ability to support large loads at small deflections with minimum space requirements in the direction of loading. They are useful in protective construction applications requiring limited shock attenuation and as backup systems to reduce shock in the even of bottoming of coil springs. They are relatively inexpensive and readily available in capacities of up to about 60,000 lb. Changes in loading conditions are accommodated by the addition or removal of units. Variations in thickness and elastic limit of commercial grade springs may cause the spring rate to vary by as much as ±20 percent, although special-grade springs may be fabricated to maintain the spring rate.

(c) The steels normally used for the smaller sized springs include AISI C1074, C1085 and C1095. Low-alloy steels such as silico-manganese steel AISI 9260 or AISI 6150 are preferred for the larger springs. When the loadings are primarily static, the maximum allowable working stresses may be taken equal to the compressive elastic limit of the material. Under these conditions, local yielding may occur around the top inner edge of the spring, resulting in a redistribution of the stresses and a more even distribution of load throughout the spring. The springs may be stacked in parallel, in series, or in series-parallel, to obtain a wide variety of responses. For greater deflection, hardened steel spacer rings may be placed between the discs. A higher load for a given deflection is obtained from the parallel arrangement. This arrangement is less predictable than the series or individual arrangements because of friction damping and breakaway friction. The series arrangement provides greater deflection for the same load but increases the probability of instability.

(d) For the very heavy duty springs, the maximum axial deflections are limited, by allowable stress, to values less than the thickness of the metal. The disc spring elements could be used in the flat form, making flat plate theory apply in analysis. In practice, however, it is probably economical to forge the spring elements to the shallow conical configuration commonly described as coned disc (or Belleville) springs (fig. 6-30). In this form, if the deformation height, h, is made equal to the maximum allowable deflection, δ, under load, the disc elements can be stacked to provide the necessary axial deflection clearance without use of spacer rings. Maximum allowable deflection per disc is analogous to the maximum allowable deflection per turn in a helical coil spring, and the required total deflection must be provided by stacking the necessary number of discs in series. Spring rate per isolator assembly is then equal to the disc spring rate divided by

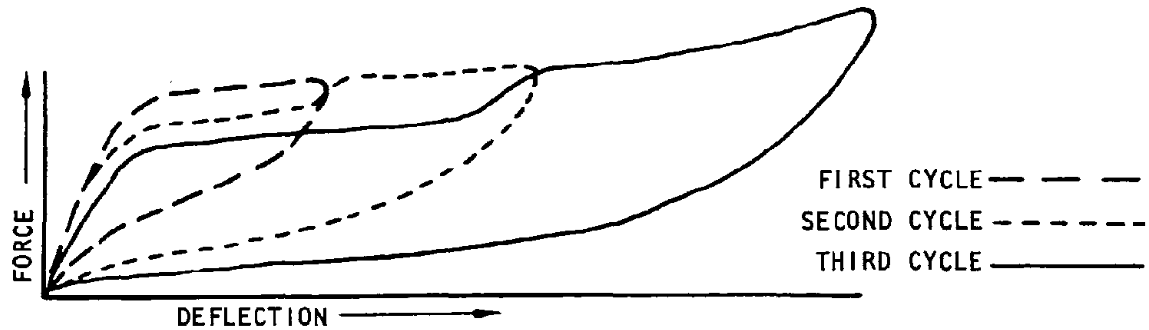


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FIGURE 6-26. EFFECT OF PREWORKING FOAM TO OBTAIN STABILIZED PROPERTIES

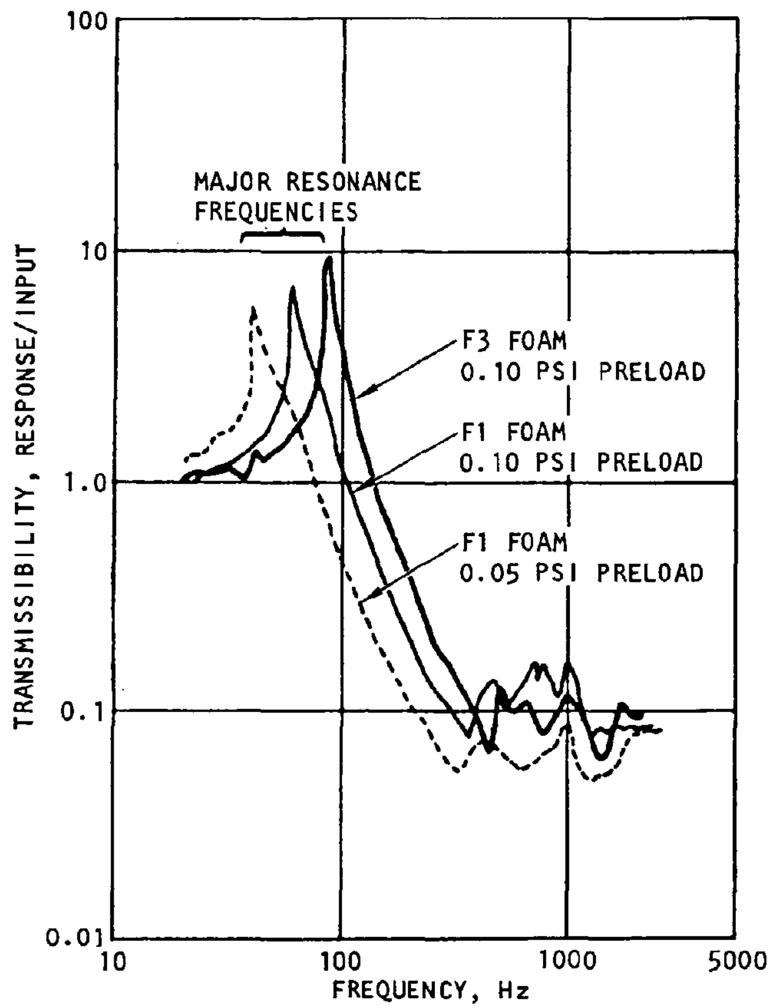

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FIGURE 6-27. TYPICAL FOAM TRANSMISSIBILITY-TENSION-COMPRESSION

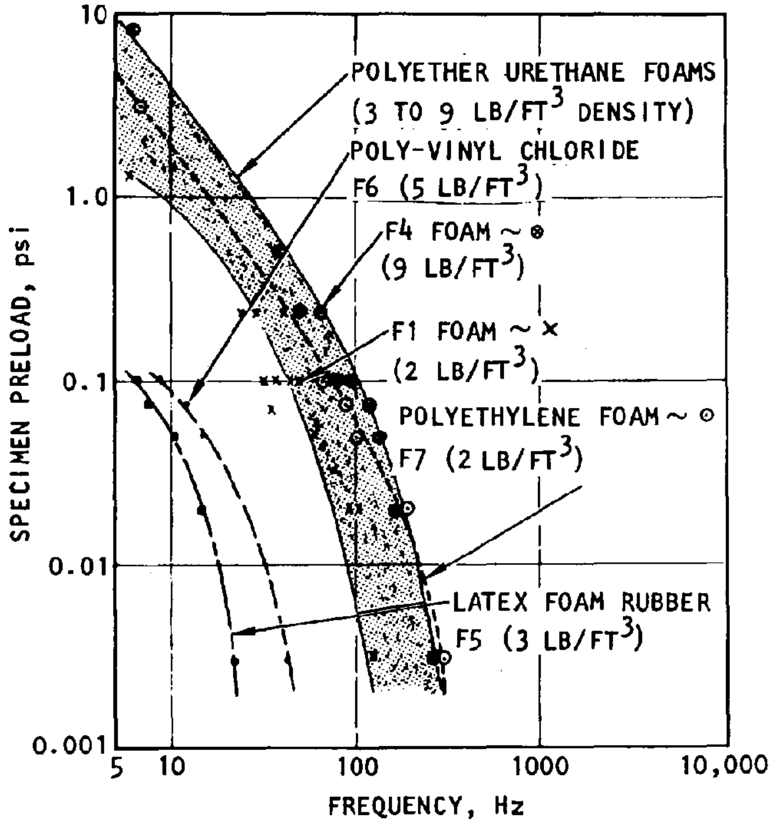

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FIGURE 6-28. FOAM RESONANT FREQUENCIES VERSUS PRELOAD

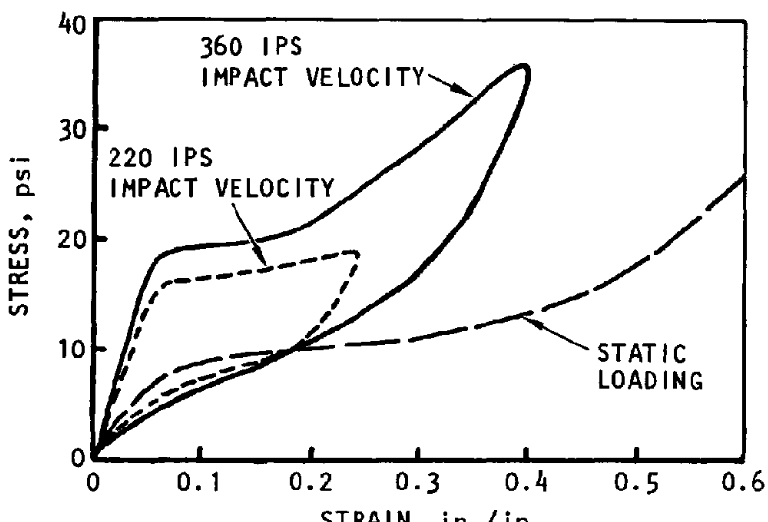

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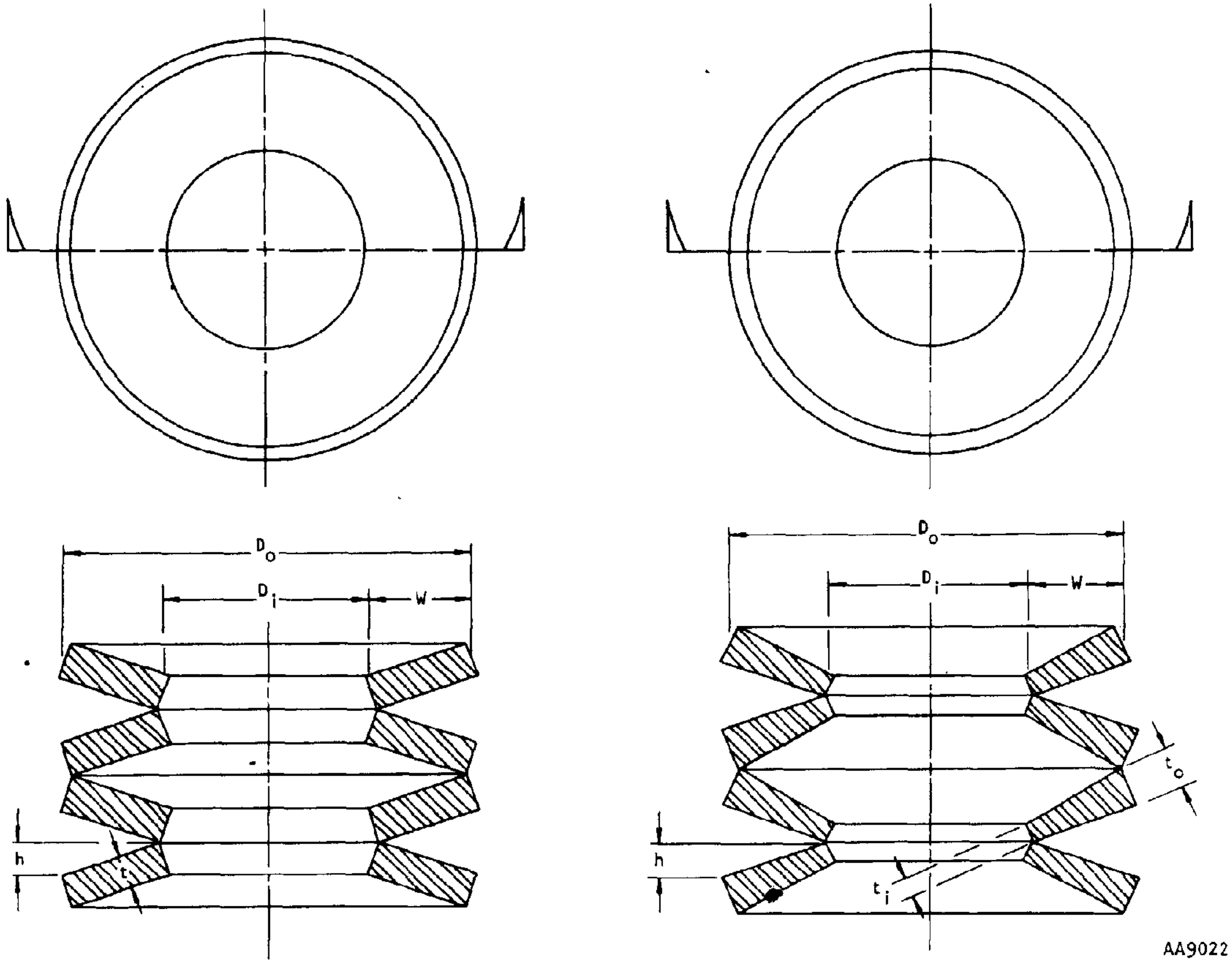
FIGURE 6-29. COMPARISON OF POLYURETHANE FOAM STRESS/STRAIN RELATIONSHIPS FOR DYNAMIC AND STATIC COMPRESSION LOADING



(a) Coned-disc springs

(b) Radially tapered (and coned) disc springs

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Figure 6-30. Disc Springs

the number of discs per isolator.

(e) A heavy-duty, double-acting, coned-disc isolator is shown in figure 6-31. Units of this type have a total load capacity of up to 10^6 lb and spring rates on the order of 15,000 to 25,000 lb/in.

(f) Coned-disc springs possess many attractive features, but as yet have not been used in protective structures. Failure of one or more discs in the device only reduces capacity, in contrast to the failure of a helical spring. At maximum stroke, spring rates increase substantially (Schnorr-Neise, n.d.), thus reducing the bottoming impact. Design details covering disc springs may be obtained from Schnorr-Neise (n.d.), Wahl (1963), and SAE (1971).

(3) Pneumatic bellows.

(a) The air bag and air bellows are two common configurations for pneumatic bellows both function in the same general manner. As shown in figure 6-32, the air bag consists of a single section subject to shape and volume changes, whereas the air bellows has two or more sections. Both types are usually constructed of fabric-reinforced-rubber sections assembled between steel end plates. The bellows type of construction uses one or more annular steel reinforcing rings to divide the axial length of the bag into several sections and increase the practical ratios of length and deflection to diameter. The rolling bellows is actually a positive sealing device sometimes used to replace the usual ring type of piston seal in air cylinders. The device is shown schematically in figure 6-33. The fold in the sleeve moves in the annular space between the piston wall and cylinder wall during relative motion between these parts.

(b) Pneumatic bellows have very nonlinear spring characteristics; however, the spring rate tends to become more linear as the amount of compression increases. They are self-restoring and not strain-rate sensitive. Although they do not exhibit any significant amount of damping, damping can be provided by the addition of reservoirs to the system. The recommended maximum static operating pressure is usually 100 psi. These springs must be inspected regularly to ensure that the proper static height is being maintained. In the case of closed systems, the spring cannot be adjusted to compensate for changes in static loading.

(c) The design of pneumatic bellows is not as simple as that of other springs, since much of the design data are empirical and determined experimentally for each spring by the manufacturer. Prior to designing a pneumatic bellows spring, a data sheet, or plot of the volume, effective area, and load versus deflection, must be obtained from the manufacturer of the spring.

(d) The usual design criteria for a pneumatic bellows spring include the static load, the maximum spring rate, and the maximum dynamic displacement. With these criteria and the manufacturer's data sheets, the required characteristics of the spring system can be determined. For any specific spring and set of initial conditions i.e., spring height, internal volume and pressure, and load the dynamic load versus deflection relationships can be determined from the characteristic curves and elementary gas laws. The total available displacement should be twice the specified dynamic displacement plus a safety margin of about 20 percent.

(e) Pneumatic bellows have not been used to date in shock isolation for protective structures, but may be considered for special applications to protect fragile subsystems.

(4) Constant force isolators.

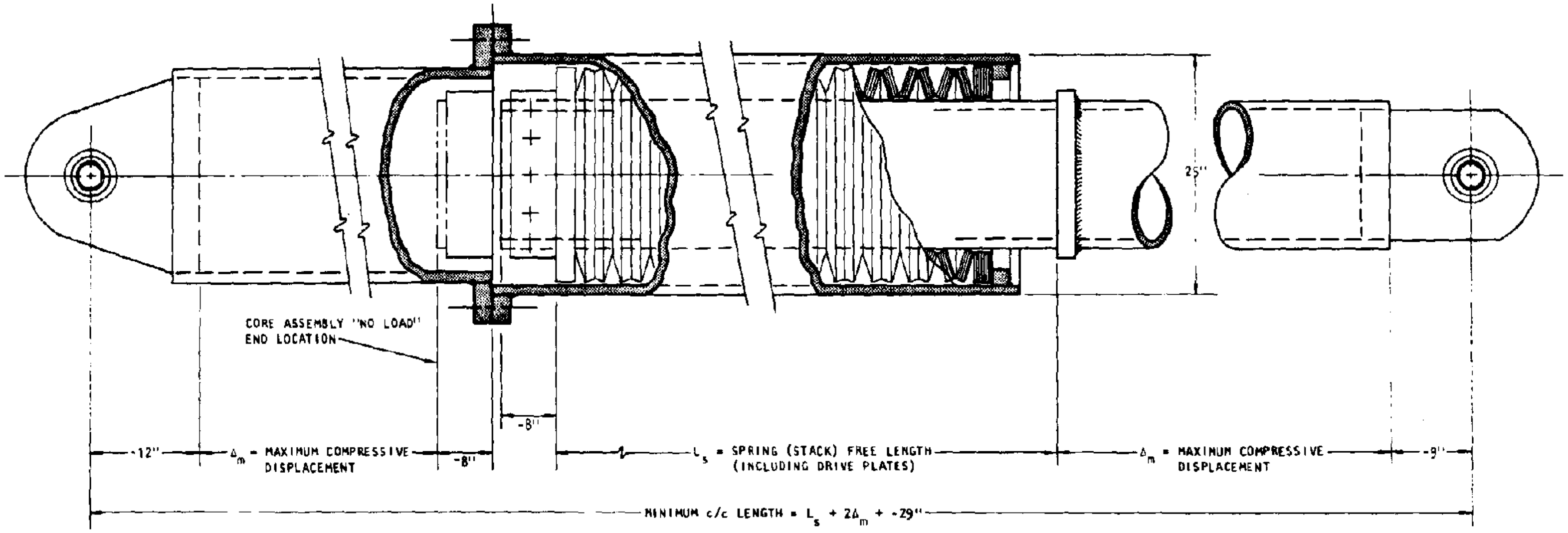
(a) Cyclic plastic straining of a ductile metal in a fixed strain range produces a hysteresis loop that stabilizes during the first few cycles. Repeated cycling results in almost constant energy absorption per cycle until eventual fatigue failure. Metals such as aluminum can tolerate roughly 1000 such cycles before failure, while stainless steel or titanium will approach 10,000 cycles before a fatigue failure (Platus et al., 1973).

(b) The narrower the hysteresis loop, the greater the cycles to failure and the greater the total energy absorption at failure. Cyclic plastic straining to failure results in considerably greater energy absorption than unidirectional straining to failure. Specific Energy Absorption (SEA) for N cycles to failure is approximately $\sqrt{N}$ times the SEA for unidirectional straining to failure. These principles are illustrated qualitatively in figure 6-34.

(c) One type of constant force isolator is shown schematically in figure 6-35. In this type of device, the ring or tube elements are compressed slightly out-of-round into the plastic range. Stroking of the device causes the elements to roll and produces cyclic plastic bending deformation as the points of maximum strain are translated around a ring or tube. To ensure that the elements roll rather than slide, the ratio of squeeze force to roll force is selected so that the sliding friction force is always greater than the roll force, i.e., the coefficient of friction times the squeeze force is greater than the roll force. It is desirable, however, to maintain the squeeze force only as high as necessary to minimize the tube wall thickness and the device weight. In another variation of this device, the rolling element is a torus rather than separate rings.

(d) These units will not return an isolation system to its original equilibrium position. Final platform displacement and angle are determined upon removal of dynamic loads.

(e) These units have not been applied to a protective-structure isolation system, although their dynamic characteristics lend themselves to optimum



U.S. Army Corps of Engineers

AAB961

Figure 6-31. Heavy-Duty, Double-Acting Strut: Capacity to 1,000,000 Maximum Load Using Coned-Disc Springs (Typical Spring Rate 15,000 lb/in to 25,000 lb/in)

(a) Air bag spring (b) Air bellows spring

Figure 6-32. Typical Pneumatic Bellows Configurations (AJA, 1966)

shock isolation (Platus et al., 1973; Platus, 1973). An excellent application is to connect these units in series with a conventional isolator to circumvent bottom-out situations, particularly where strong probabilistic nuclear threat conditions exist. This series system is illustrated in figure 6-36.

(5) Solid elastomer springs.

(a) "Elastomeric" and "rubberlike" are terms that are often used to describe rubber or similar materials. The term "polymer" is used to designate rubber in the raw or uncompounded state. Springs made from these materials are often called "shock mounts" because of their wide use in shock isolation applications. They are normally used in medium-to-light-duty applications and represent an economical solution to the isolation of small items of equipment. These springs are fabricated from a wide variety of natural and synthetic rubbers and compounds and in numerous sizes and shapes, satisfying a wide range of applications. Because of the range in capacity and characteristics of commercially available units, only in unusual cases is it necessary to design a unit. The larger units are usually custom designed for specific applications.

(b) In most applications, the solid elastomer spring requires little space and exhibits good weight-to-energy storage ratios. In addition to load and deflection, the operating environment should be considered in selecting an elastomer for a specific application. The desirable properties of some elastomers can be significantly degraded when exposed to low or high temperatures, sunlight, ozone, water, or petroleum products.

(c) The response of elastomeric springs is nonlinear in most applications because of the nonlinear stress/strain properties of elastomers. The springs are self-damping because of the viscoelastic properties of the elastomers. They are almost always in compression because of bonding limitations, although rubber has been used in torsion spring applications for the Navy Bureau of Ships as hinges for bow ramps. Harris and Crede (1976), Crawford et al. (1973), and Snowden (1968) contain guidance for the design of elastomeric springs.

6-3. Active isolator elements.

a. Active vibration control systems can be constructed that show better performance than the best possible passive systems or can accomplish tasks not possible by passive means. Active systems in general are more costly, more complex, and therefore less reliable than passive systems. To date, active means of shock and vibration control have been limited to cases where performance gains outweigh the disadvantages of increased cost, complexity, and weight. In protective facilities, no active shock-isolation systems have been employed up to this time.

b. Typical active isolator mechanisms include servomotor-actuated mechanical linkages, variable resilience devices containing conductive or magnetic fluids, and pneumatic or hydraulic valve-operated actuators. The active isolator mechanisms are power-operated in accordance with command signals derived

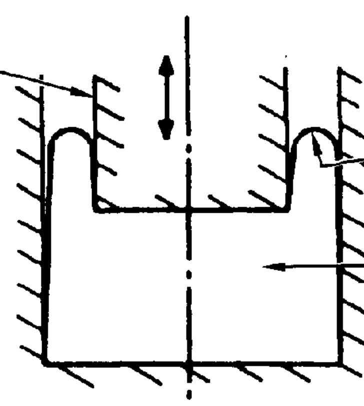


Figure 6-33. Rolling Bellows (Sleeve) Configuration (AJA, 1966)

from feedback control signals.

c. Active isolation of shock and vibration involves the application of automatic feedback control techniques to achieve the major performance objectives of a very stiff system for constant applied loads (static force or mass loading and sustained acceleration) and a very soft system for oscillatory dynamic excitations. Active isolation systems are servo-mechanisms comprising excitation and/or response sensors, sensor signal processors, and actuators. The sensors provide signals proportional to dynamic excitation or response quantities. The signal processors modify and combine sensor signals to create a command signal. The actuators apply forces or induce motion in accordance with the command signal.

d. A wide variety of excitation and response sensors can be employed to provide feedback signals to form a closed-loop control system. For example, feedback signals can be developed that are a function of jerk, acceleration, velocity, displacement, integral displacement, differential pressure, or force. The signal processor may consist of an active electronic network that performs amplification, attenuation, differentiation, integration, addition, and compensation functions. Alternatively, the signal processor can be in the form of a simple lever mechanism.

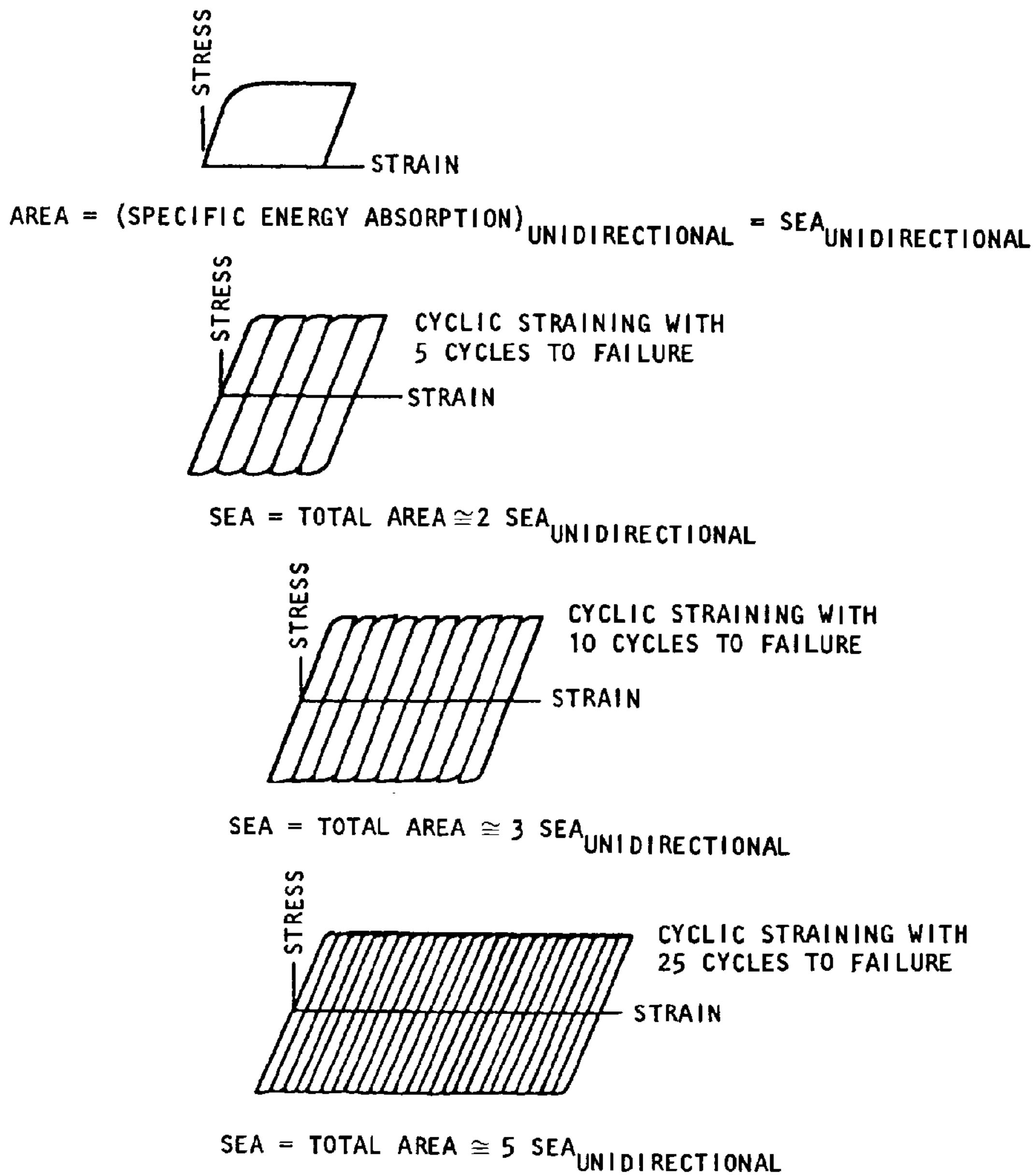


Figure 6-34. Relation between Hysteresis Loop and Specific Energy Absorption (Platus, 1973)

e. Power is required to operate the active isolator mechanisms and, in some cases, is also required for the signal processor. The requirement of externally supplied power is the primary distinguishing characteristic between active and passive isolation systems. A major consideration in the design of active isolation systems is the achievement of an adequate margin of stability while providing a high speed of response.

f. Of the active isolator mechanisms, only the mechanopneumatic isolator and the electrohydraulic isolator have so far been developed to the status of practical, stable operational systems (Ruzicka, 1967).

(1) Mechanopneumatic isolation systems (Ruzicka, 1968). A schematic diagram of a mechanopneumatic isolation system is shown in figure 6-37. The system employs a double-acting pneumatic actuator; mechanical displacement feedback controls the flow of a compressible gas to and from the actuator through a servovalve. The effect of displacement feedback actuation of the valve spool is to make the actuator output force a function of the time integral of relative displacement. With integral displacement control, no static deflection results from mass loading and transient deflections due to sustained acceleration are initially reduced and eventually eliminated.

(a) Integral displacement control can be designed to be effective only at extremely low frequencies so the isolation system's natural frequency will not be affected materially. Consequently, vibration and shock isolation are provided essentially in accordance with the stiffness and damping characteristics of the passive pneumatic actuator.

(b) Isolation system damping may be provided by an external damping mechanism or a flow-restriction damping device inserted between the pneumatic actuator and surge tank, as illustrated in figure 6-37. Surge tanks allow lower natural frequencies to be achieved, provide excellent resonant vibration control, and offer a 12 db/octave high-frequency attenuation rate regardless of the degree of isolator damping used.

(c) The vibration transmissibility of the mechanopneumatic isolation system with surge tanks is shown in figure 6-38. Zero and infinite damping are achieved by providing no restriction and infinite restriction, respectively, of the cyclic flow of gas between the actuator and the surge tanks. An optimum degree of damping is required to minimize the resonant transmissibility of the isolation system. However, optimum damping is not critical, since a fairly large deviation from optimum damping results in a relatively small increase in resonant transmissibility. When a capillary flow-restriction damping device is used, the vibration-isolation characteristics of the mechanopneumatic isolation system (fig. 6-38) are similar to those exhibited by a linear passive isolation system employing an elastically coupled viscous damper.

(d) Since the bandwidth of active control for the mechanopneumatic isolation system is relatively narrow, the speed of response is generally limited and the servocontrol system may be incapable of following the high-frequency components of a given shock excitation. Integral displacement feedback tends to decrease the relative displacement to a magnitude lower than that exhibited by a passive isolation system having an equal natural frequency, and improved control of dynamic deflections under shock can be achieved by increasing the feedback gain. However, since an increase in integral displacement feedback gain is equivalent to a reduction in system damping, such action would be detrimental to resonant vibration control and system stability. The hardening passive-stiffness characteris-

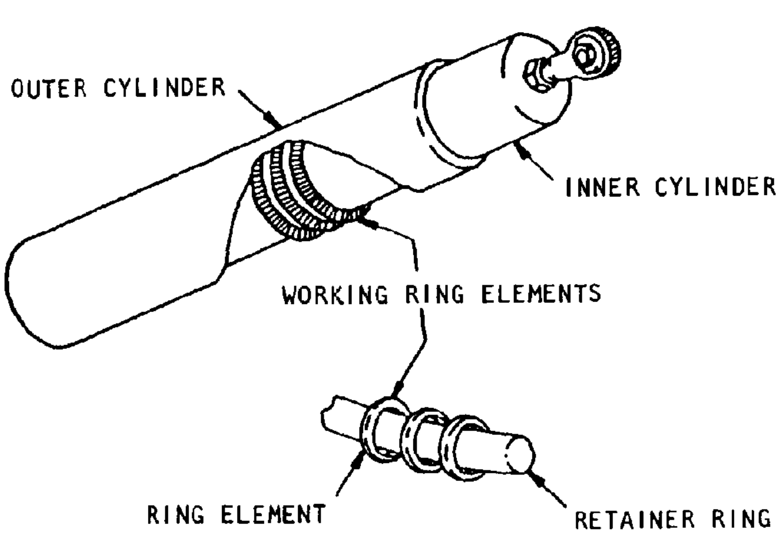
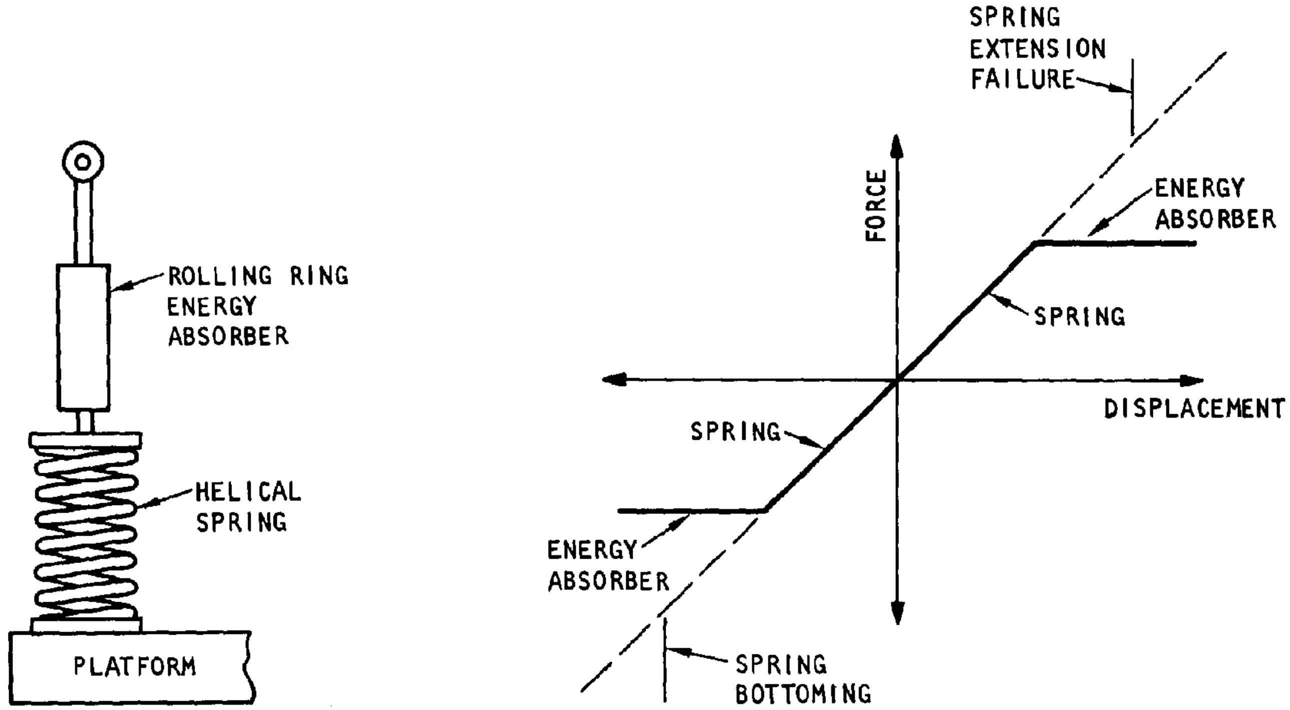


Figure 6-35. Rolling Ring Energy Absorber (Platus, 1968)



(a) Series-connected spring isolator and rolling ring energy absorber

(b) Force displacement of spring-energy absorber

Figure 6-36. Series-Connected Spring Isolator and Rolling Ring Energy Absorber to Prevent Spring Overloads (Platus et al., 1973) tic of pneumatic actuators generally is advantageous in reducing the dynamic deflection under shock excitation.

(e) Typical transient response characteristics of the mechanopneumatic isolation system subjected to acceleration-step excitation are shown in figure 6-39. The acceleration response is essentially that of a conventional passive system with an equal natural frequency. However, the magnitude of relative displacement is reduced below that of a comparable passive isolation system and eventually becomes zero. Because of the integral displacement control, the isolated body ultimately is returned to its neutral position; isolation of vibration can be provided during conditions of sustained acceleration.

(f) Having eliminated the static deflection and reduced the dynamic displacement by use of integral displacement control, it becomes possible to construct mechanopneumatic isolation systems having natural frequencies less than conventional passive systems. Specifically, natural frequencies in the range of 0.5 to 20 Hz can be provided in practical installations. Since the isolator stiffness can be varied automatically in proportion to the payload weight by using integral displacement feedback to achieve height control, the isolation system performance is relatively insensitive to variations in payload weight. However, because of the compressibility of the gas and the necessity of having relatively low gain in the feedback loop to ensure system stability and good resonant vibration control, the speed of response of the mechanopneumatic isolation system is relatively slow. Furthermore, since the effective stiffness characteristic is passive and bilateral, low-frequency mechanopneumatic isolation systems are responsive to dynamic excitation imposed on the isolated mass from sources such as payload dynamics, acoustic pressure variations, and personnel movements. Finally, the physcial size and weight of the lower natural frequency systems may become objectionable in some applications.

(2) Electrohydraulic isolation systems (Ruzicka, 1968). A schematic diagram of an electrohydraulic isolation system is shown in figure 6-40. Multiple electronic feedback signals are processed through a servoamplifier to create a command signal that controls the flow of a relatively incompressible fluid to and from a cylinder through a servovalve. Sensors are employed to provide acceleration and relative displacement feedback signals, which are modified in the servoamplifier. The flow through the valve is made a function of acceleration, relative velocity, relative displacement, and the time-integral of relative displacement. The effect of each feedback parameter may be independently controlled by adjusting its gain.

(a) The stiffness of the passive hydraulic actuator is high, which provides a high natural frequency system in the open loop. Upon closing the loop, extremely low natural frequencies can be provided, at values substantially lower than 1 Hz, since the isolation system resonance is created electronically. Typical vibration transmissibility characteristics of the electrohydraulic isolation system with various combinations of feedback parameters are shown in figure 6-41. These are applicable in the low-frequency region, where the effect of the sensor and hydraulic component dynamics is negligible. For high-frequency vibration, the sensor and hydraulic component dynamics cannot be neglected, and compensation is generally required to ensure a stable closed-loop system.

(b) Pure acceleration ($\ddot{x}$) feedback provides a resonance-free transmissibility characteristic that is a unit lag function $1/(\tau s + 1)$, where the time constant $\tau - C_a/A$, C_a is the acceleration feedback flow gain, A is the effective actuator piston area, and s is the Laplace operator. The combination of acceleration ($\ddot{x}$) and relative displacement (δ) feedback provides a response

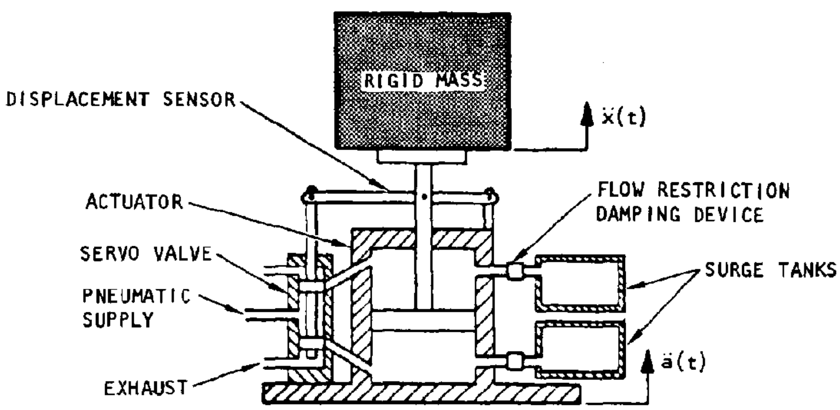


Figure 6-37. Schematic Diagram of Mechanopneumatic Isolation System Employing Double-Acting Pneumatic Actuator and Surge Tank Damping Mechanism (Ruzicka, 1968)

identical to a conventional viscous damped passive isolation system. In this case, the natural frequency (in Hz) is given by

$$f_o = \frac{\omega_o}{2\pi} = \frac{1}{2\pi}\sqrt{C_d/C_a} \qquad (6\text{-}5)$$

and the effective viscous damping ratio $\zeta = C/C_c$ is

$$\zeta = \frac{A}{2\sqrt{C_a C_d}} \qquad (6\text{-}6)$$

where C_d is the relative displacement flow gain.

(c) For acceleration ($\ddot{x}$), relative velocity ($\dot{\delta}$) and relative displacement (δ) feedback, the response is again identical to a conventional viscous damped passive isolation system. In this case, the natural frequency is given above and the effective viscous damping ratio ζ is

$$\zeta = \frac{C_v + A}{2\sqrt{C_a C_d}} \qquad (6\text{-}7)$$

where C_v is the relative velocity flow gain.

(d) Hence, the addition of relative velocity ($\dot{\delta}$) feedback provides a means of increasing damping without affecting the natural frequency of the isolation system. Finally, the effect of integral displacement ($\int \delta dt$) feedback can be introduced (by use of lead compensation in the acceleration feedback loop, for example) without materially affecting the high-frequency isolation characteristics.

(e) The transmissibility characteristics of the electrohydraulic isolation system shown in figure 6-41 are applicable over a limited frequency range. Degradation of high-frequency vibration isolation occurs because of the effects of hydraulic resonances, which are determined by the dynamic characteristics of the hydraulic components and the electronic networks. This is illustrated by the "active" transmissibility curve in figure 6-42, which indicates a low isolation-system natural frequency (created electronically by selecting appropriate values of feedback gains C_a and C_d), and a hydraulic resonance in the high-frequency region (typically in the range of 50 to 150 Hz). The transmissibility generally exceeds a value of unity at the hydraulic resonance and, for higher frequencies, the isolation characteristics are those of a passive isolation system having a natural frequency associated with the hydraulic resonance condition.

(f) Improved high-frequency isolation can be

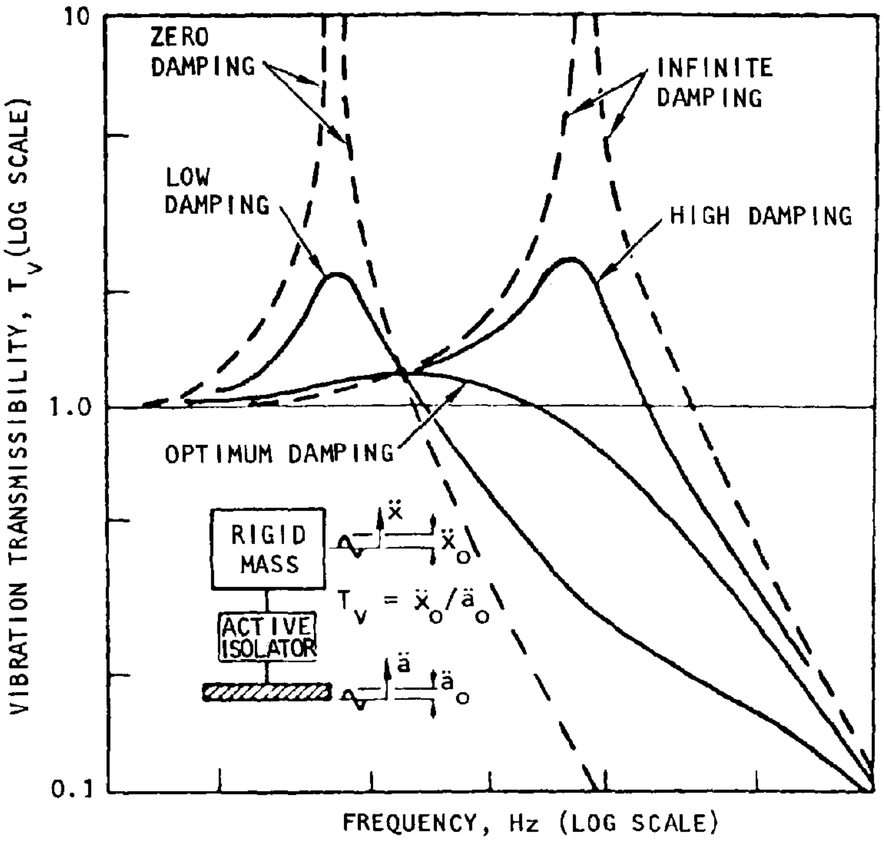
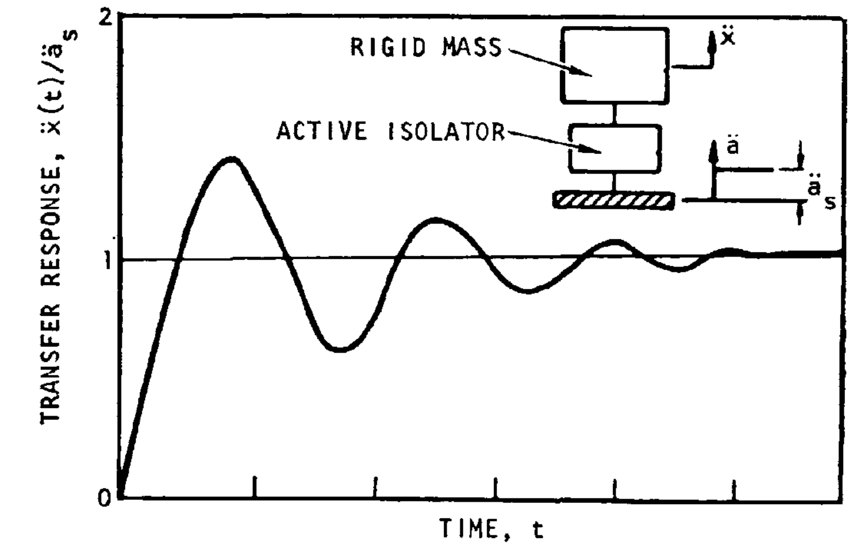


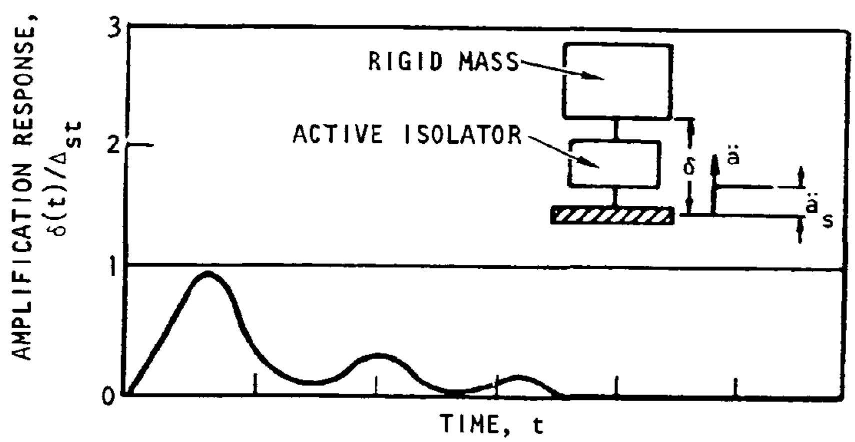
Figure 6-38. Vibration Transmissibility Characteristics of Mechanopneumatic Isolation System with Surge Tank Damping Mechanism (Ruzicka, 1968).

provided by introducing a suitable passive isolator in the form of a flexible coupling in the servo loop, as illustrated earlier in figure 6-41. The effect of the flexible coupling is to provide mechanical compensation, which results in reshaping the high-frequency response characteristics. The isolator mechanism then provides broad-band isolation, as indicated by the "active-passive" transmissibility curve shown in figure 6-42. The active mechanism isolates vibration in the low-frequency region where the passive mechanism has a unity transfer function. The passive mechanism provides isolation of high-frequency vibration, where the isolated body is effectively decoupled from the hydraulic actuator and the excitation frequency is beyond the frequency band over which the servocontrol system is operative.

(g) With proper selection of feedback gains, the linear system will exhibit the desired position control and transmissibility characteristics. In response to shock excitation, however, large dynamic deflections of linear electrohydraulic isolation systems may result because of the extremely low natural frequencies that can be achieved. If nonlinear electronic compensation is introduced into the acceleration or relative-displacement feedback loop, the loop gain changes substantially for relative displacements exceeding an established linear range of operation. This has the effect of providing a hardening stiffness characteristic with greatly increased feedback control operating to limit the relative displacement severely and to rapidly reposition the isolator in its region of linear operation. Such nonlinear compensation will result in a higher accelera-



(a) Acceleration response for step input



(b) Relative displacement response for step input

Figure 6-39. Transient Response Characteristics of Mechanopneumatic Isolation System for Acceleration Step Excitation (Ruzicka, 1968).

tion. However, the isolation system will respond as a stiff system for only a limited time duration (because of the automatic and rapid repositioning of the system to its linear range of operation), after which the full degree of vibration isolation is restored.

(h) Typical transient response characteristics of the electrohydraulic isolation system subjected to acceleration step excitation are shown in figure 6-43. Pure acceleration feedback ($\ddot{x}$) provides excellent control of acceleration but requires an infinite relative displacement. As discussed previously, the combination of acceleration ($\ddot{x}$) and relative displacement (δ) feedback provides performance comparable to a conventional passive isolation system, with the capability of electronically creating an extremely low natural frequency. Additional system damping can be provided by use of relative velocity feedback. By introducing integral feedback, excellent control of both acceleration and relative displacement response is provided. A high speed of response in eliminating the relative displacement provides vibration isolation during conditions of sustained acceleration. Nonlinear electronic compensation can be used to provide even greater reduction of relative displacements.

(i) As an alternative to acceleration feedback, the force transmitted by the isolator can be used as a primary sensor feedback signal. Comparable isolation characteristics are provided by use of either acceleration or force feedback; however, different isolator dynamic stiffness characteristics will result. For acceleration feedback, the isolator stiffness is unilateral; that is, a very low stiffness is presented to the dynamic environment but a high stiffness is presented to forces acting on the isolated mass. Consequently, electrohydraulic isolation systems with acceleration feedback are relatively insensitive to payload dynamics. For force feedback in lieu of acceleration feedback, however, the isolator dynamic stiffness is bilateral; that is, the isolator presents a very low stiffness to both the environment and payload, and the isolation system is consequently sensitive to payload dynamics and changes in payload weight.

(j) The feedback gains for acceleration C_a and relative displacement C_d determine the natural frequency (in a manner similar to the mass m and stiffness k in a passive system). The ability to select feedback gains C_a and C_d over a very wide range (compared to the limited ranges for the mass and stiffness counterparts of these parameters in a passive isolation system) makes it possible to provide extremely low natural frequencies of electrohydraulic isolation systems that are independent of payload weight. Electronic compensation, however, introduces the possibility of instability, thereby limiting the range of permissible feedback gains. Natural frequencies in the range of 0.01 to 10 Hz can be provided in practical installations, and even lower natural frequencies are feasible.

(k) Because of the wide selection of feedback signals, loop gains, and compensation schemes available, ultralow-frequency vibration isolation can be provided even during conditions of sustained acceleration, with zero static deflection, a high speed of response,

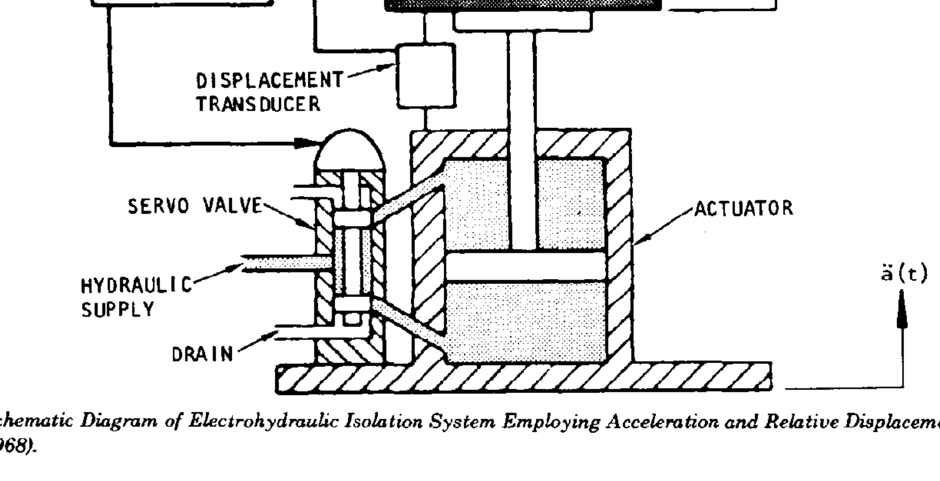


Figure 6-40. Schematic Diagram of Electrohydraulic Isolation System Employing Acceleration and Relative Displacement Feedback (Ruzika, 1968).

and extreme flexibility in shaping the overall frequency response characteristics. The size and weight of the electrohydraulic isolator is practically invariant with natural frequency, but could become objectionable for smaller payloads.

(3) Summary of natural frequency ranges for mechanopneumatic and electrohydraulic isolators.

(a) Based on experience with actual hardware systems, a summary of the ranges of practical application of passive and active isolation systems is presented in figure 6-44. The range of equivalent passive static deflection associated with the natural frequencies included in the comparison is also given.

(b) Passive isolators provide natural frequencies in the range of 3 Hz and above, with static deflections that vary with the inverse square of the natural frequencies.

(c) Mechanopneumatic isolators can provide natural frequencies ranging from 0.5 to 20 Hz. Mechanopneumatic isolators are sometimes referred to as semiactive isolators, since relatively low-gain servocontrol is employed to maintain a constant height of isolator. Nevertheless, isolation is provided essentially in accordance with the passive dynamic characteristic of the pneumatic actuator.

(d) Electrohydraulic isolators can provide natural frequencies over the range 0.01 to 20 Hz, with even lower natural frequencies considered feasible. For natural frequencies less than approximately 0.045 Hz, comparable passive isolation systems would exhibit deflections measured in miles.

g. In response to shock excitation, large dynamic displacements of linear isolation systems may occur because of the extremely low natural frequencies that can be achieved. Consequently, it is generally desirable to harden the effective stiffness characteristics. This is accomplished automatically in the mechanopneumatic isolation system, since the isolator stiffness has a hardening characteristic that is described approximately by a quadratic force-deflection characteristic. For the electrohydraulic isolation system, nonlinear electronic circuits can be introduced into the displacement feedback loop to substantially increase the loop gain for relative displacements exceeding an established linear range of operation. This has the effect of providing a hardening stiffness characteristic with greatly increased feedback control operating to rapidly reposition the isolator in its region of linear operation.

h. The hardening stiffness and feedback effects of the active isolation systems result in a higher accelera-

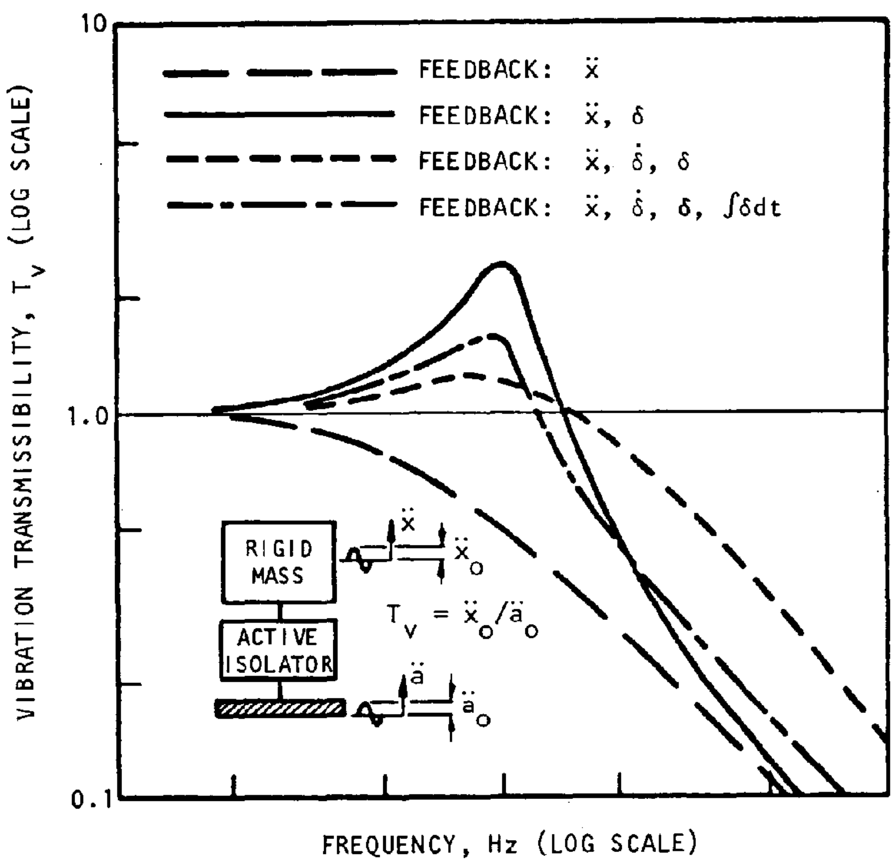


Figure 6-41. Low-Frequency Vibration Transmissibility Characteristics of Electrohydraulic Isolation System with Various Combinations of Feedback Parameters (Ruzicka, 1968).

tion being transmitted to the isolated body during severe shock excitation. For the electrohydraulic isolation system, however, the variety of feedback parameters provides considerably more flexibility than do passive isolation systems in achieving an acceptable compromise between the acceleration and relative displacement response maxima.

i. Applications of active system can be found in Ruzicka (1968) for an inertial guidance platform (fig. 6-45) and in Calcaterra and Schubert (1968) for a pilot's seat (fig. 6-46). Calcaterra and Schubert (1968) and Bies and Yang (1968) provide examples for helicopters.

6-4. Semiactive isolators elements (Crosby-Karnopp, 1973; Karnopp et al., 1974).

a. Recognizing both the performance benefits as well as the limitations of active isolation elements, designers developed a new concept in isolation. This concept involves the application of a controllable force generator that does not require significant external power. Called an active damper, this device produces a controllable force that is derived from the relative velocity of its attachment points. Like the externally powered active element, it can produce forces that are functions of measured system variables. The variables are appropriately combined to form a command signal for the device. Since the device does not utilize significant external power, its performance is more limited than that of the fully active force generator.

b. The active damper is a new concept in the control of shock and vibration. It provides system performance intermediate between that of a passive isolation system and of a fully active isolation system. The active damper is an externally controllable force generator where the force is developed by the relative velocity of its attachment points. Essentially a controllable energy dissipator, it does not require significant external power for its operation. Like the fully active system, its output force is controlled as a function of measured system variables such as velocities and accelerations.

c. During portions of its operation where the command force and available force are in opposite directions, the active damper is controlled to produce essen-

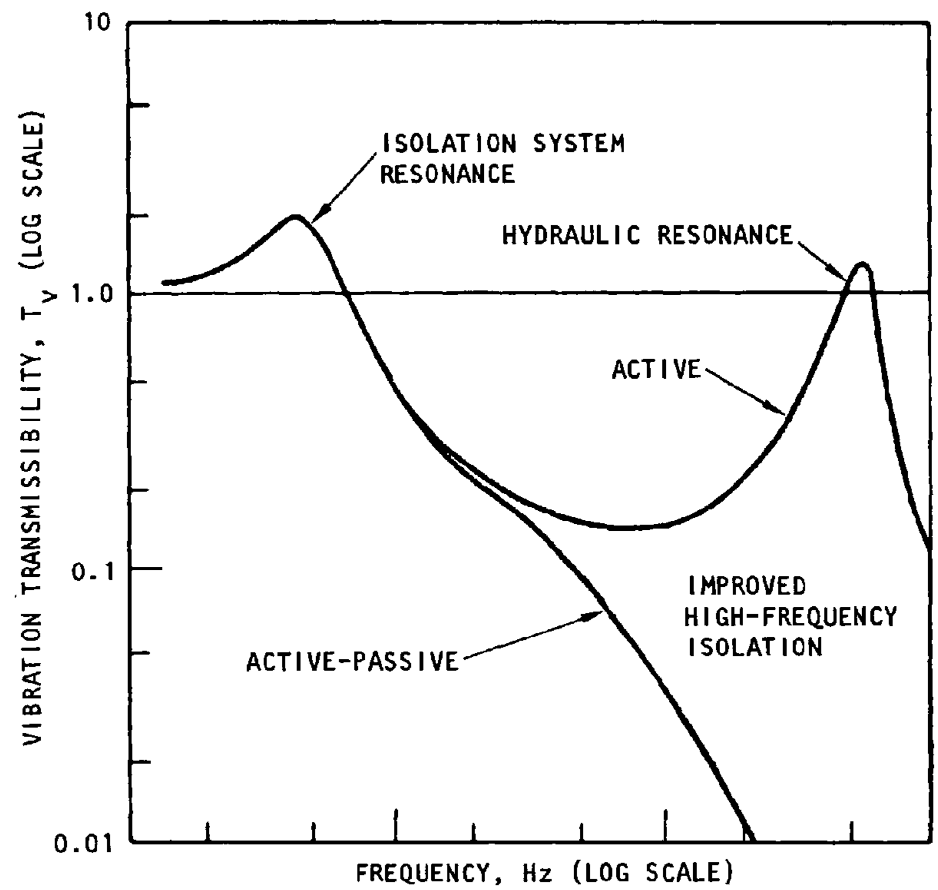


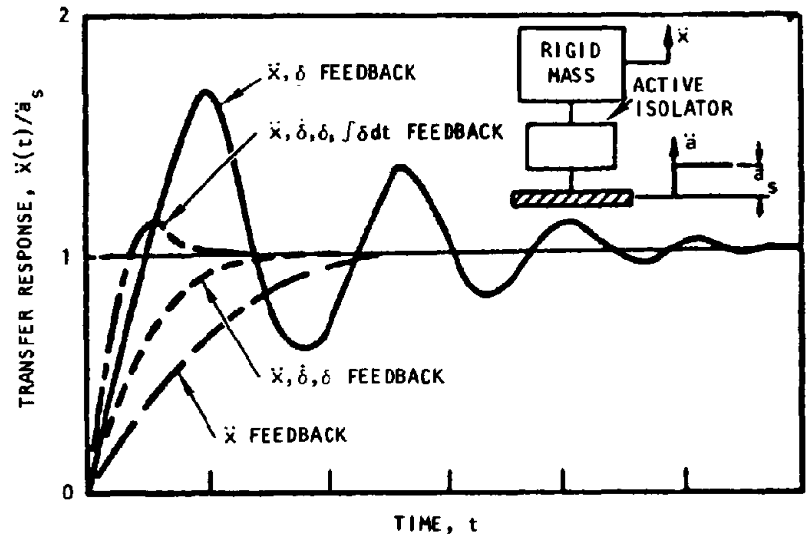
Figure 6-42. Vibration Transmissibility Characteristics of Electro-Hydraulic Isolation System Demonstrating Use of Passive Isolator in the Servo Loop to Improve High Frequency Isolation (Ruzicka, 1968).

tially zero force. If the command force exceeds the available force, the active damper experiences a lock-up mode, which transmits the maximum available force. Both of these conditions degrade system performance relative to a fully active system.

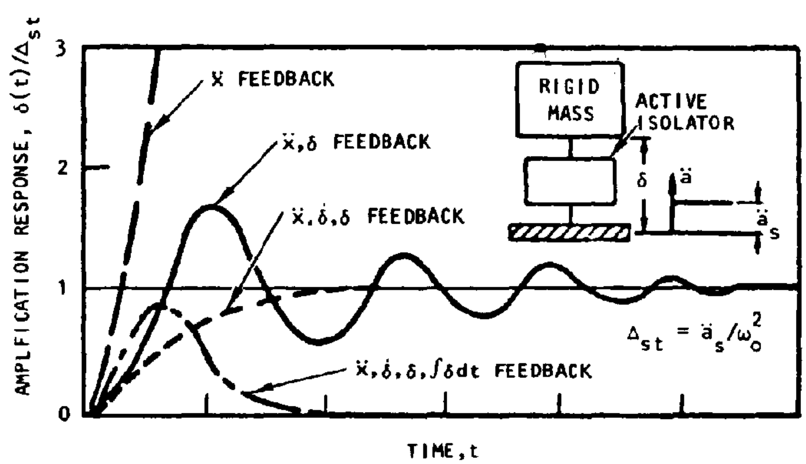
d. There are several ways in which a semiactive force generator might be realized in practice. Figure 6-47 shows a schematic diagram of an active damper that resembles a conventional direct acting hydraulic shock absorber, except that the hydraulic pressure and hence the force is controlled by a pair of poppet valves.

(1) The force required to open the valves is set by a torque motor or a staged force amplifier. As figure 6-47 illustrates, one valve sets the compressive force and the other the tensile force. When a compressive force is commanded, the compression valve sets the force when the damper is being compressed. The tensile force valve is unloaded and acts like a check valve, assuring that when the damper is extending, the tension force will be virtually nil. Similarly, when a tensile force is commanded, the tension valve sets the force when the damper is extending. The compression valve is unloaded so that virtually no compression force arises if the damper is compressed.

(2) The torque motor or other force amplifier requires a small power supply, but the larger damper force is generated in a passive manner. Hence the device is called "semiactive" and functions as a force and



(a) Acceleration response for step input



(b) Relative displacement response for step input

Figure 6-43. Transient Response Characteristics of Electrohydraulic Isolation System with Various Combinations of Feedback Parameters for Accelerations Step Excitation (Ruzicka, 1968).

power amplifier. The result is that a small amount of active power controls the vibration through the modulation of a larger amount of dissipative power.

(3) The semiactive system requires sensors such as accelerometers and relative velocity transducers and a control unit to command the valve force actuator. However, the sensors operate at signal power level and the actuator at lower power. A fully active system would require, in addition, a high-power actuator with good frequency response; this would entail a relatively large power supply and a high-power, fast servomechanism. Clearly, any dissipation device that can be modulated can form the basis for a semiactive system.

e. Experimental results using standard industrial valves (not fig. 6-47) are given in figure 6-48.

6-5. Shock-isolated platforms, equipment, weapons.

a. High quality isolators are only a part of the solution in reducing facility shock to equipment and personnel. Isolators may be used to support platforms, may be directly connected to equipment, and may serve as mounts to weapons. The dynamic characteristics of the platforms, equipment, and weapons interact with the isolators and may significantly alter transmission characteristics (figs. 5-13 to 5-22 in chap. 5).

b. To attenuate facility shock motions, it is critical to reduce interaction of the isolated load. Interaction is normally reduced by ensuring adequate stiffness in the platforms and equipments. The general rule is to require that first-mode resonance of the isolated load be approximately four times the resonance of the load on its isolation system. For medium-sized systems, this is not an unreasonable requirement. For large systems, it has been found that the first torsion or bending modes are below the equivalent "rigid" body resonance.

c. Platforms must be designed for stiffness in bending and in torsion, which will often require numerous deep "I" beams with both top and bottom plates (figs. 8-3, 8-4). Equipment mounted on platforms can be braced at their bases; the braces provide further stiffening by acting as doublers or hat sections. Equipment should be mounted into the platform beams and not into the cover plate. Wherever possible, equipment cabinets should be bolted together to provide further stiffening and improve friction energy loss (Fischer, 1977). For telephone-rack type of equipment, bridge-types of structures will be required.

d. The range of parameters for shock-isolated platforms used in the Safeguard ABM system is given in the table 6-2.

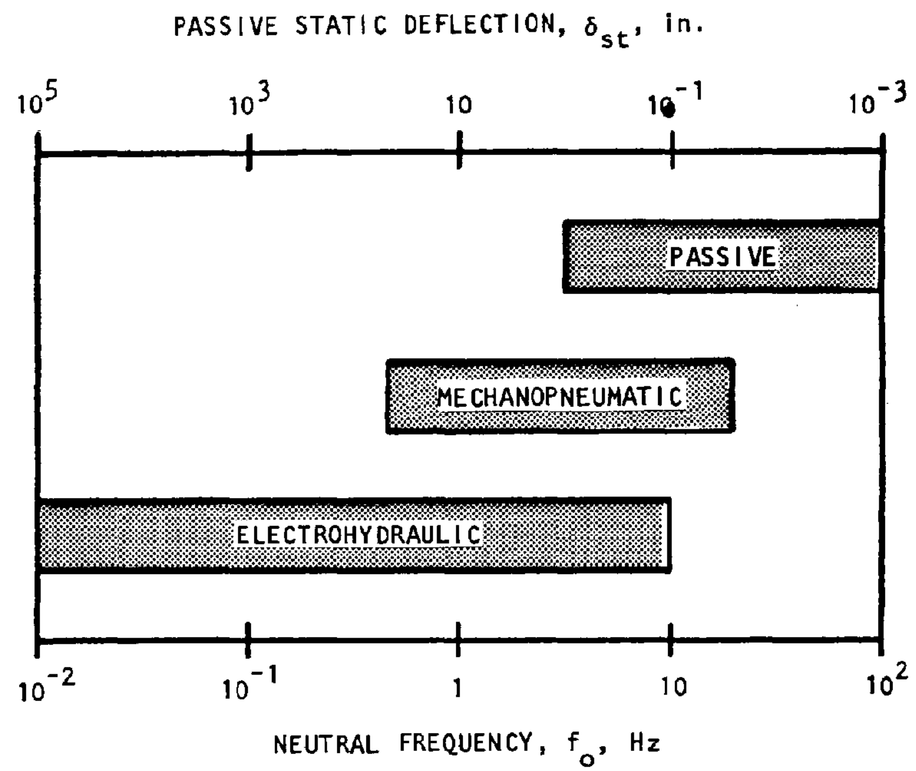


Figure 6-44. Practical Ranges of Natural Frequency of Passive, Mechanopneumatic, and Electrohydraulic Isolation Systems (Ruzika, 1968).

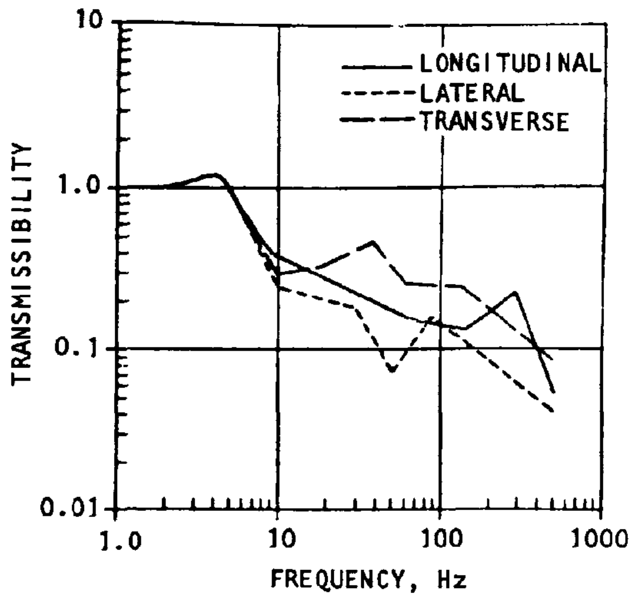


FIGURE 6-45. TRANSMISSIBILITY CHARACTERISTICS OF MECHANOPNEUMATIC ISOLATION SYSTEM FOR MISSILE INERTIAL GUIDANCE STABLE PLATFORM (Ruzicka, 1968)

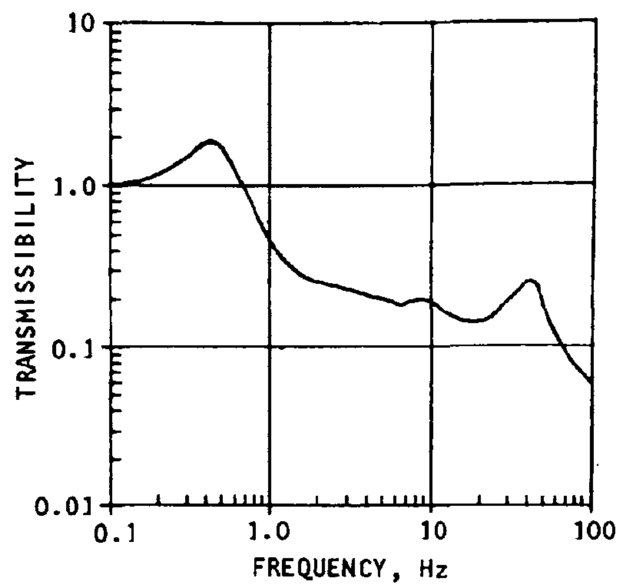


FIGURE 6-46. TRANSMISSIBILITY CHARACTERISTICS OF ELECTROHYDRAULIC ISOIATION SYSTEM FOR JET AIRCRAFT PILOT SEAT (Calcaterra-Schubert, 1969)

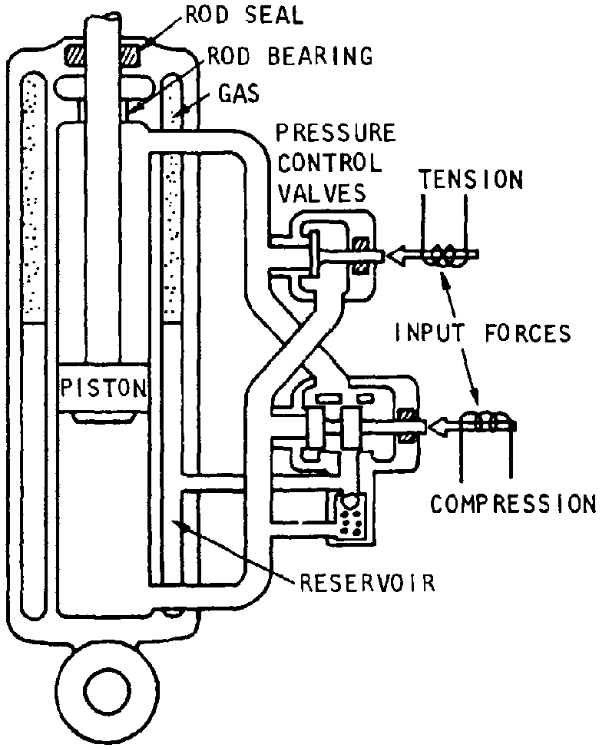


Figure 6-47. Electrohydraulic Active Damper Schematic (Karnopp et al., 1974)

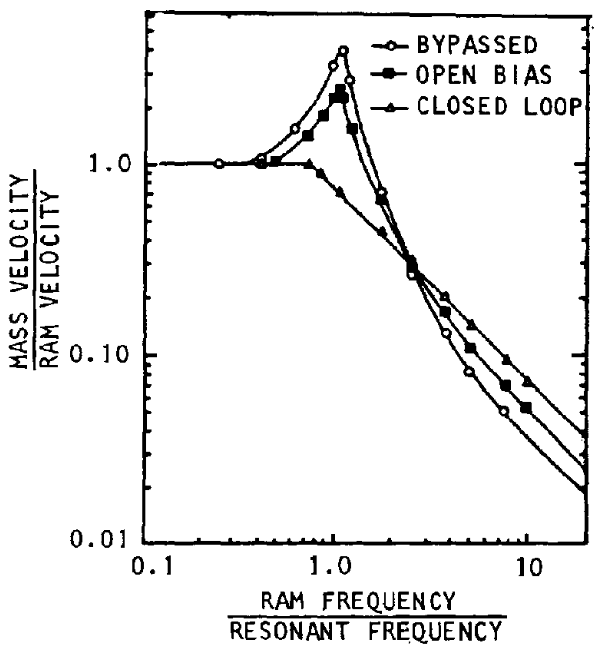


Figure 6-48. Experimental Results for Breadboard Semiactive Isolator (Karnopp et al., 1974).

TABLE 6-2. SHOCK ISOLATION SYSTEM PARAMETERS (Boyd-Huang, 1977)

	Range	
	Minimum	Maximum
Platform Length, ft	4.5	77.0
Platform Width, ft	3.0	50.0
Aspect Ratio, L/W	1.0	15.4
Platform Area, sq/ft	15	3,136
Total Weight, lb	1,400	284,000
Platform Weight, lb	400	132,000
Weight Ratio (total wt/plat. wt)	1.5	11.8
Density (total wt/area), lb/ft^2	59	647
Moment of Inertia, $in.^4$	40	20,095
Frequency (Modal), Hz	0.6	91.7
Isolators (Numbers)	4	60
Total Weight/Isolator, lb	350	18,050
Isolator Static Loads, lb	50	20,000
Isolator Stiffness, lb/in.	5	1,675
Isolator Natural Frequency, Hz	0.1	2.0
Isolator Stroke, in.	± 4	± 8
Isolator Damping, % of critical	0	20

e. Vertical transfer inertance functions $[\ddot{x}/F(\omega)]$ across two SAFEGUARD platforms are shown in figures 6-49 and 6-50 (Safford-Walker, 1975). These data are reasonably representative of the platforms measured. For figure 6-49, switchgear cabinets (not connected to each other) were mounted on a rigid platform having diagonal beams and a bottom shear panel. For figure 6-50, control equipment, consoles, and personnel were mounted in box frame with paralell beams and open bottom. It is to be noted that the inertance of the smaller platform is 25 times that of the larger one.

f. A systematic study (experimental and analytical) of the application of damping materials to shock-isolated platforms was conducted by C. F. Vail (1972).

(1) Damping materials were Class II Navy tile, MIL-P-23653; 3M viscoelastic material SJ20003X; and spaced constrained layer damping, Lord Mfg. Co.

(2) A stiff and a flexible isolated platform were used in this study (see fig. 6-51). The span distance of 11 ft was chosen as a typical separation distance between isolators.

(3) The experimental test arrangement is illustrated in figure 6-52. The three 1000-lb equipment cabinets mounted on the platform were specifically designed for high structural integrity forces in protective facilities. Excitation by the electrodynamic shaker was a constant amplitude slow sine sweep (1 oct/min), 10 to 2000 Hz. The transient test for the same frequency band was a rapid sine sweep at two amplitude levels for somewhat over 1 sec duration each.

(4) The several configurations of damping treatments to the shock isolated floor are shown in figures 6-53 and 6-54.

(5) Mode shape data were used to calculate the inherent damping of the specimen from both high level and low level transfer function amplitudes. The results are plotted in figure 6-55. It was found that the modes did not fully develop at the lower input level due to the resisting inertia of the cabinets. Since the theoretical

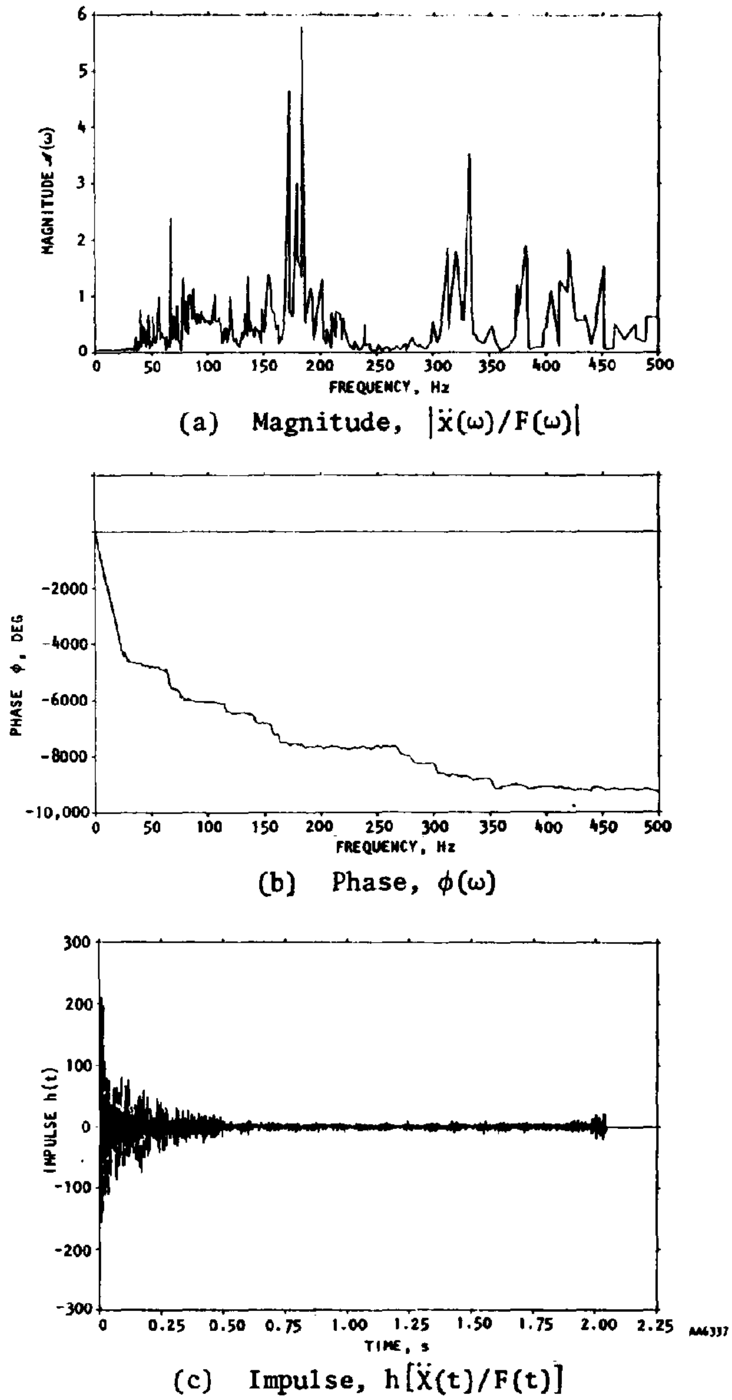
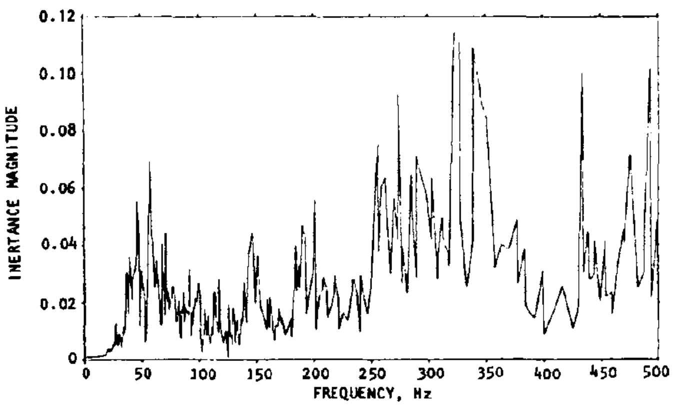
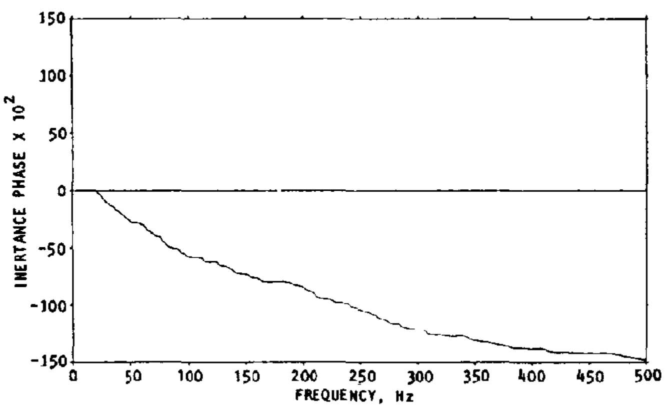


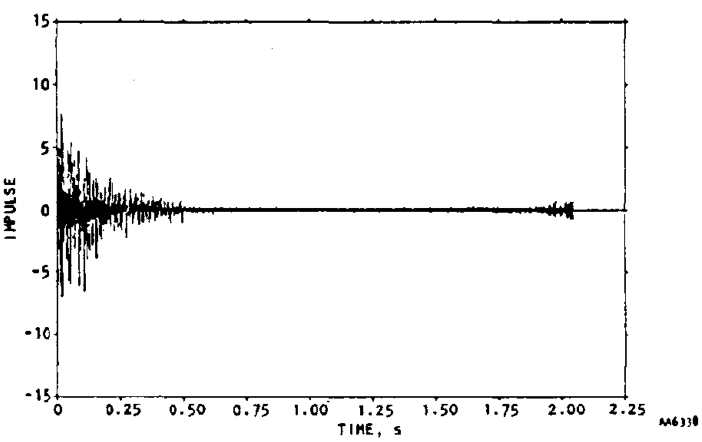
Figure 6-49. Vertical Transfer Inertance Function [$\ddot{x}/F(\omega)$] across Platform from Isolator Attachment Point to Mid-Platform (Platform Characteristics: 23,500 lb, 23 ft by 7 ft, 6 Pneumatic Isolators Viscous Damped) (Safford-Walker, 1975).



(a) Magnitude, $|\ddot{x}(\omega)/F(\omega)|$



(b) Phase, $\phi(\omega)$



(c) Impulse, $h[\ddot{X}(t)/F(t)]$

Figure 6-50. Vertical Transfer Inertance Function [$\ddot{x}/F(\omega)$)] across Platform from Isolator Attachment Point to Mid-Platform (Platform Characteristics: 198,000 lb, 64 ft x 49 ft, 18 Helical Coil Isolators with External Friction Dampers) (Safford-Walker, 1975).

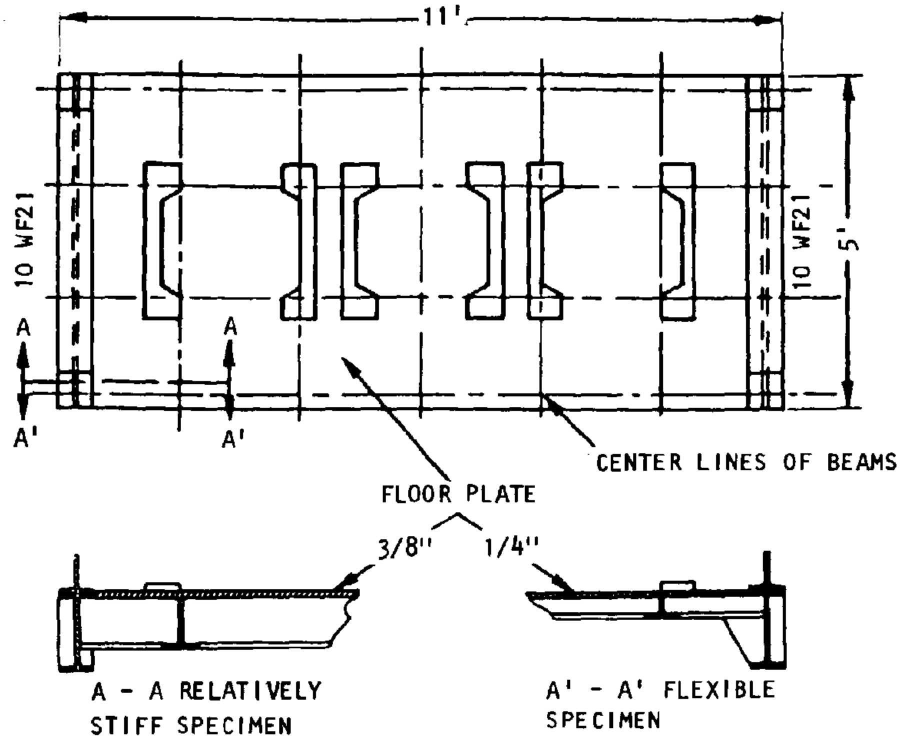


Figure 6-51. Schematic of Floor Test Specimens (Vail, 1972)

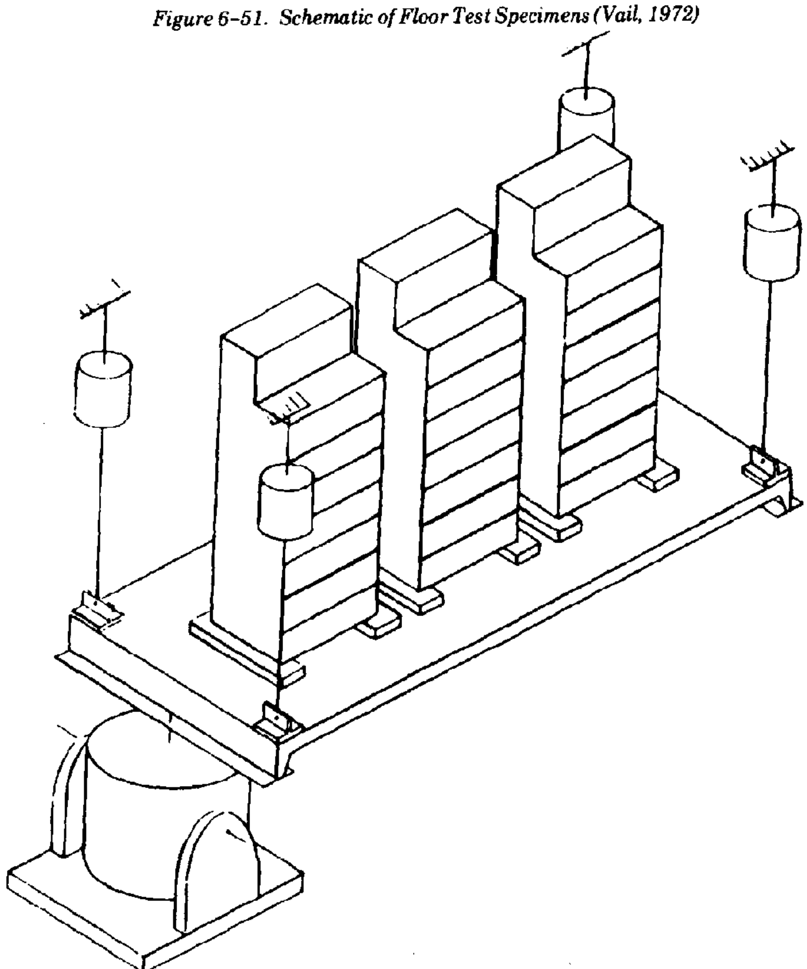
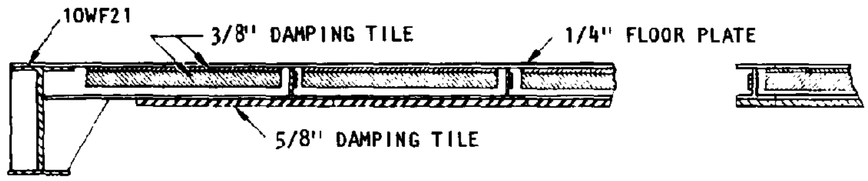
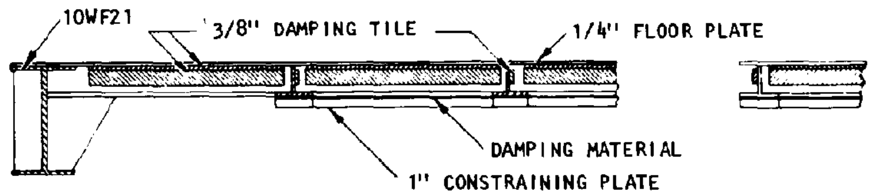


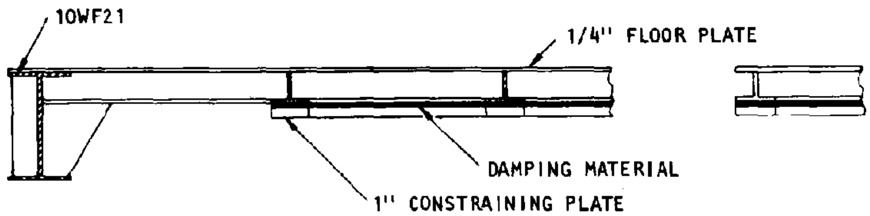
Figure 6-52. Schematic of Test Arrangement (Vail, 1972)



(a) Free layer additive damping

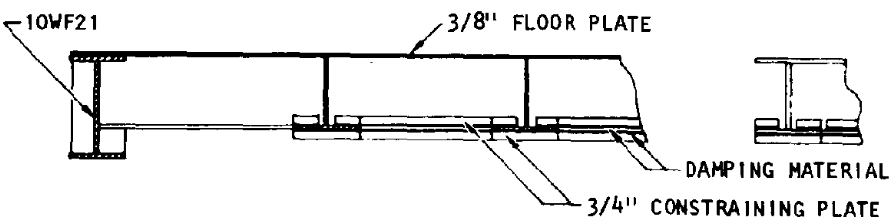


(b) Free layer and constrained layer additive damping

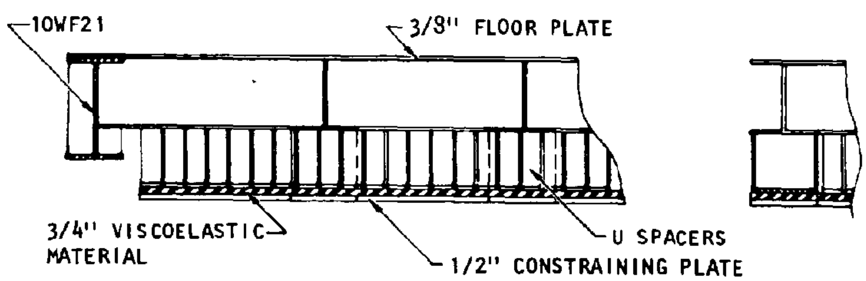


(c) Constrained layer additive damping

Figure 6-53. Additive Damping Configurations for Flexible Steel Specimen (Vail, 1972)



(a) Constrained liner



(b) Spaced constrained liner

Figure 6-54. Relatively Stiff Steel Floor Additive Damping Configurations (Vail, 1972)

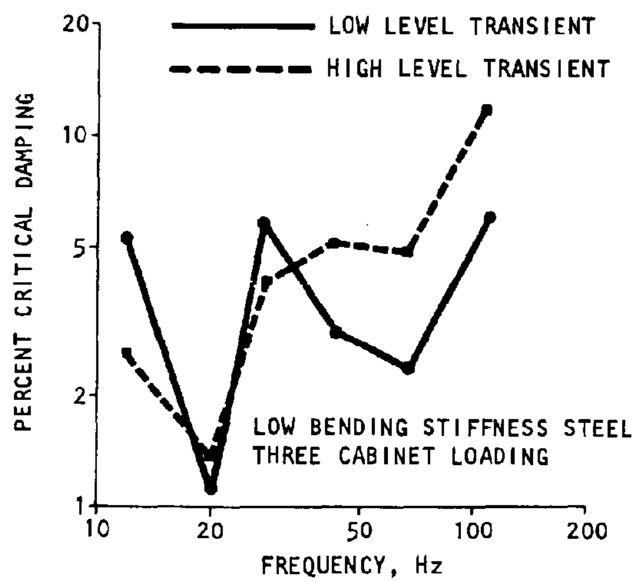


FIGURE 6-55. COMPARISON OF CALCULATED INHERENT DAMPING FROM LOW-LEVEL AND HIGH-LEVEL TRANSIENT (Vail, 1972)

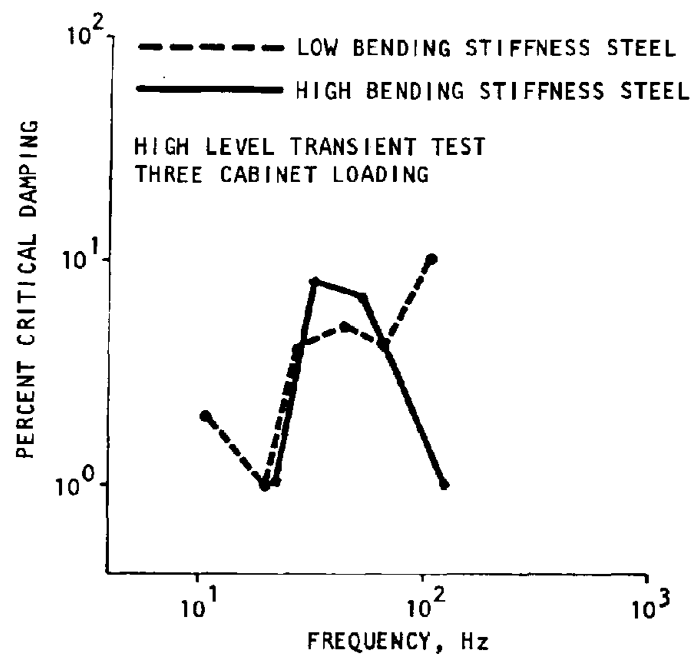


FIGURE 6-56. CALCULATED INHERENT DAMPING OF TEST SPECIMENS (Vail, 1972)

modes were used to work back to find the inherent damping and modal development was amplitude dependent, it is not surprising that results for the lower input level indicated higher damping below 30 Hz. A possible explanation for higher damping with higher input amplitude at frequencies 30 Hz was the activation of stick-slip friction forces.

(6) A comparison between the calculated inherent damping for the low bending stiffness specimen and high bending stiffness specimen is shown in figure 6-56. Note that high damping existed in the frequency range 20-80 Hz, corresponding to the range where large cabinet displacements occurred. There are many interface frictional sources in the cabinet, such as the back panel and plenum panels, which could cause the large damping in this frequency range. Interfacial slip-

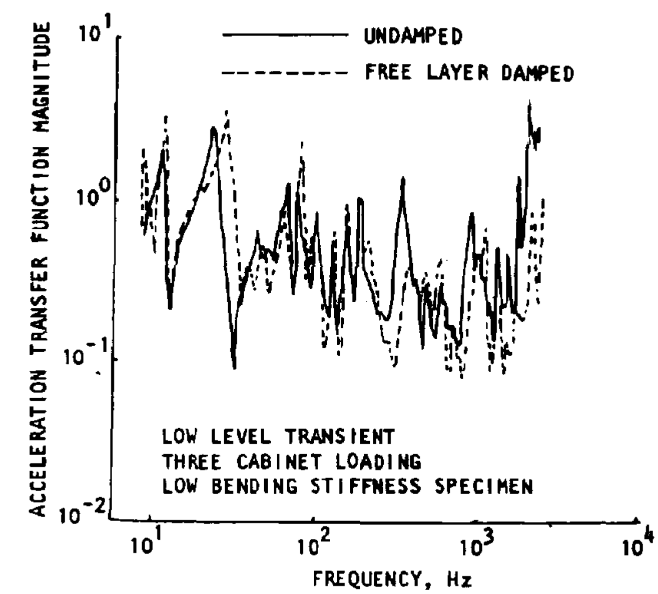


FIGURE 6-57. COMPARISON OF LOW BENDING STIFFNESS SPECIMEN TRANSFER FUNCTION MAGNITUDES UNDAMPED TO FREE LAYER DAMPED AT CABINET BASE NEAR INPUT (Vail, 1972)

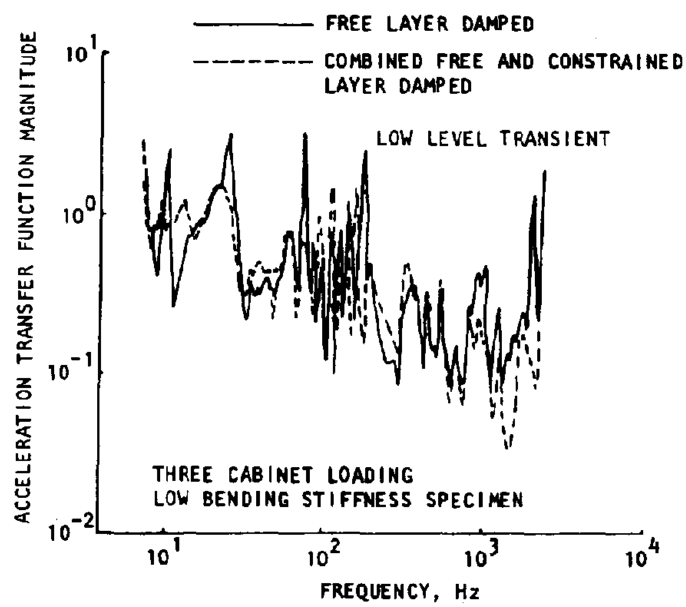


FIGURE 6-58. COMPARISON OF LOW BENDING STIFFNESS SPECIMEN TRANSFER FUNCTION MAGNITUDE FREE LAYER DAMPED TO COMBINED FREE LAYER AND CONSTRAINED LAYER DAMPED AT CABINET BASE NEAR INPUT (Vail, 1972)

page made clear impressions at the cabinet foot and mounting pad attachment locations.

(7) Effects of damping treatments on the shock-isolated floor are presented in transfer function format in figures 6-57 through 6-61. Damping treatments are compared to each other and against an undamped platform.

(8) Vail's conclusions regarding the suitability of damping applications to shock isolated floors are listed below (Vail, 1972):

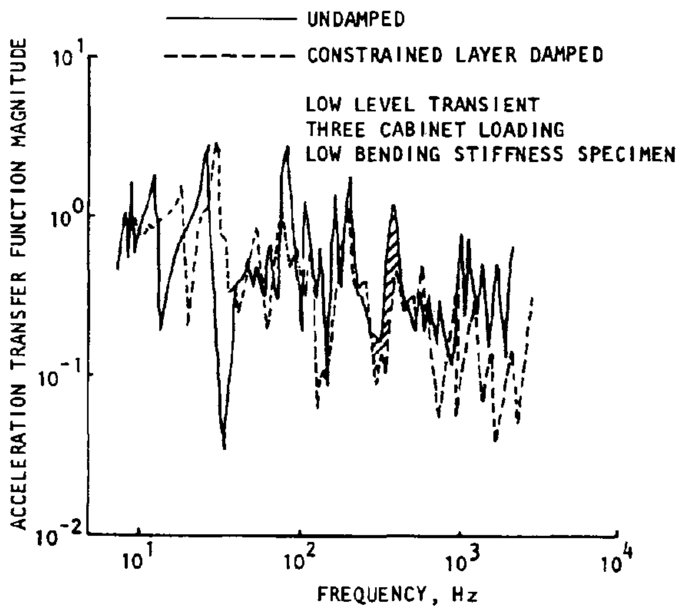


FIGURE 6-59. COMPARISON OF LOW BENDING STIFFNESS SPECIMEN TRANSFER FUNCTION MAGNITUDE UNDAMPED TO CONSTRAINED LAYER DAMPED AT CABINET BASE NEAR INPUT (Vail, 1972)

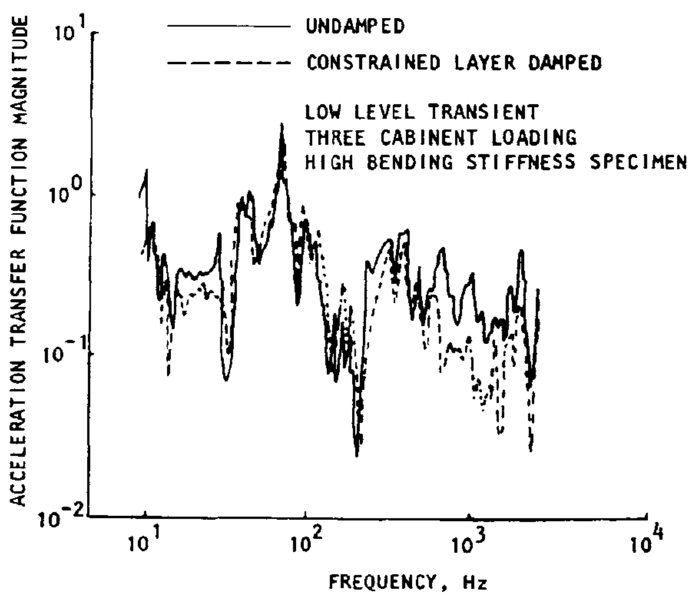


FIGURE 6-60. COMPARISON OF HIGH BENDING STIFFNESS SPECIMEN TRANSFER FUNCTION MAGNITUDE UNDAMPED TO CONSTRAINED LAYER DAMPED AT CABINET BASE FARTHEST FROM INPUT (Vail, 1972)

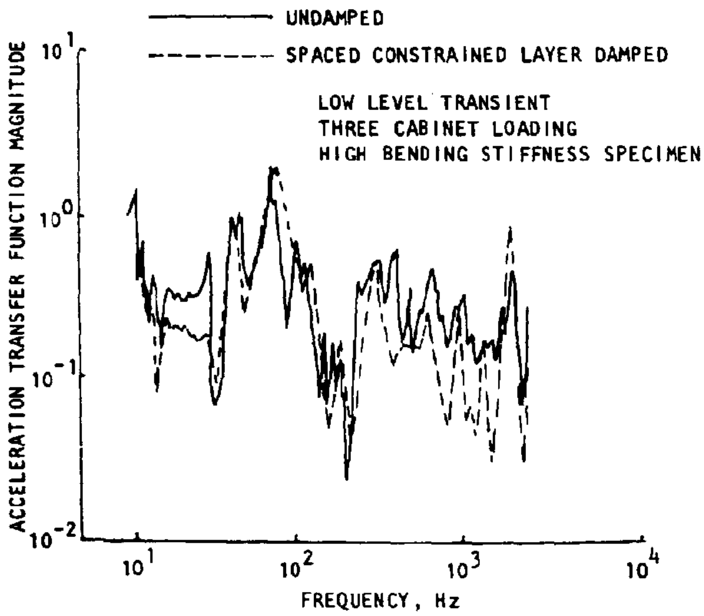


Figure 6-61. Comparison of High Bending Stiffness Specimen Transfer Function Magnitude Undamped to Spaced Constrained Layer Damped at Cabinet Base Farthest from Input (Vail, 1972)

—Inherent damping was found to be more dependent upon mode shape than upon frequency.

—Stiff specimens appeared to be far more difficult to damp than relatively flexible specimens.

—Free layer damping was ineffective in reducing the overall response of the system; however, it provided excellent attenuation of local resonant responses.

—Additive damping systems may suppress inherent damping by reducing the response and thus reducing interfacial slippage damping in the structure.

—Although additive damping almost always reduces the harmonic response, it may not reduce transient response due to the increase in area under the transfer function magnitude plot. That is, as the peaks are lowered, the valleys in the curve are filled in.

CHAPTER 7

DYNAMIC ANALYSIS

7-1. Introduction.

a. General. This chapter describes the necessary dynamic analyses that are required for shock isolation system design. These analyses include rigid body analysis, the analysis of multi-degree-of-freedom systems, finite element analysis, and the determination of rattlespace requirements. These analyses should be based on a recognition of the requirements and constraints placed on the shock isolation system. The requirements and constraints are briefly reviewed below. Several computer programs applicable to dynamic analysis are described in appendix B.

b. Requirements. A shock isolation system generally must satisfy three primary dynamic requirements:

(1) Rattlespace requirement. The displacements of the shock-isolated platform or equipment must not exceed the available rattlespace.

(2) Motion-attenuation requirement. The shock isolation system must attenuate the facility shock enough to eliminate risk of injury to personnel and damage to equipment.

(3) Motion-decay requirement. The shock isolation system must ensure that the motion of the isolated platform/equipment following a shock dies down in a short time in order that (a) the system capability to resist a second shock is not degraded and (b) operating personnel are not subjected to the unpleasant or deleterious effects of sustained vibration.

c. Constraints. Constraints imposed upon shock isolation systems are motion constraints, space constraints, and interfacing elements. Interfacing elements include hydraulic, electrical, and temperature conditioning connections to the facility.

(1) Motion constraints. Motion constraints relevant to protecting equipment and personnel by the shock isolation system have been covered in chapter 4.

(2) Space constraints.

(a) Space surrounding the shock isolation system is a critical parameter where impact with walls, floors, or adjacent equipment is to be avoided. As the rattlespace approaches zero, an optimum isolator will transmit to the isolated platform or equipment an acceleration approaching that of the base motion. On the other hand, as the rattlespace constraint approaches the facility displacement, an optimum isolator will transmit a vanishingly small acceleration (Sevin-Pilkey, 1971).

(b) For nonoptimum isolators, if resonance develops, accelerations and displacements of the isolated system can exceed base or facility accelerations and displacements. The accelerations imposed on the isolated systems by space constraints must also be less than the equipment fragility and personnel tolerances.

(c) Rattlespace limitations can require the application of nonlinear isolators, impact absorbers (foams, elastomers, hydraulic or pneumatic bumpers), constant force devices, and active and semiactive isolators.

(d) Space constraints should be minimized owing to repercussive costs caused by enlarging the facility. The amount of rattlespace required and degree of sophistication of the isolation system should be subject to a tradeoff study covering the useful life of the facility. This study should include changes in role or mission; replacement, upgrade, and addition of equipment; and changes in threat due to accuracy, yield, and sequence of attacks. For each confidence level selected, uncertainties of threat, facility response, shock isolation system transmission, and fragility must also be included.

(3) Interfacing elements. Interfacing elements possess mass, spring, and damping characteristics that are often of sufficient magnitude to substantially alter the performance characteristics of a shock isolation system. Electrical power and signal cables connecting the shock-isolated equipment and facility must be suspended with large vibration loops to minimize undesirable loading. This loading needs to be included in the performance analysis of the isolation system. Large cable loops also adversely affect facility space. Similarly, hydraulic lines, waveguides, and air conditioning ducts require flexible connections consistent with the dynamic motion of the isolated system.

7-2. Rigid body analysis.

a. General.

(1) Rigid-body analysis is essential in the evaluation of shock isolation systems, particularly in the initial development of performance, response, and system parameter characteristics in the frequency region bounding resonance. When the platform and equipment are lumped as a rigid body, rigid-body analysis can provide useful preliminary information regarding the following:

- —Resonance characteristics
- —Modal coupling
- —Relative displacement (isolator stroke and rattlespace requirements) time history and frequency response
- —Relative velocity time history and frequency response
- —Acceleration time history and frequency response
- —Stability of isolation system
- —Sensitivity of location of center of gravity and elastic centers
- —Dynamic loads on isolators, isolator attachments, and isolated equipment
- —Cable slack and cable whip in pendulum systems

(2) Using rigid-body analysis, these characteristics can be studied with relative economy. Therefore, a sufficient number of iterations can be performed to establish the general performance and specification requirements for a shock-isolated system. Based on the results of these studies, the isolation system can be optimized with respect to a performance index, design constraints, and environmental excitation (Sevin-Pilkey, 1971).

(3) An important descriptive parameter in rigid-body analysis is the number of degrees of freedom. The number of degrees of freedom of a vibrational system is the minimum number of coordinates necessary to define completely the position of the mass elements in space.

(a) A rigid body such as any of those shown in figure 7-1 can vibrate with six degrees of freedom (three translational and three rotational).

(b) An arrangement of isolators and dampers, such as shown in figure 7-1d, provides three planes of symmetry. The center of gravity of such a system can be described by six independent equations of motion. In practical applications, one or two planes of symmetry are generally found.

(c) The properties of a rigid body that are significant in dynamics and vibration analysis are the mass, the position of the center of mass (center of gravity), the moments of inertia, the products of inertia, and the directions of the principal inertial axes.

b. Linear systems.

(1) The vibration of a rigid body on resilient supporting elements and damping elements can be described under the following assumptions. The springs as resilient supporting elements are linear, having no mass or energy-dissipative characteristics. The spring constant, K, is the ratio of force, F, to relative displacement, X_r:

$$K = \frac{F}{X_r} \qquad (7\text{-}1)$$

(2) Damping elements are massless and have no spring characteristics. The damping coefficient, C, is the ratio of force, F, to relative velocity, V_r:

$$C = \frac{F}{V_r} \qquad (7\text{-}2)$$

(3) For a single-degree-of-freedom system such as that in figure 7-2, with excitation in the vertical (Z) direction, damping is generally expressed as a fraction or percentage of critical damping (C_c):

$$\zeta = \frac{C}{C_c} = \frac{C}{2\sqrt{Km}} \qquad (7\text{-}3)$$

where $C_c = 2\sqrt{Km}$ is the critical damping coefficient, and m is the mass of the resiliently supported body. C_c is the smallest value of C for which the free damped motion is nonoscillatory and ζ is the damping ratio. An alternative damping ratio is the commonly used symbol Q, sometimes called the quality factor, which is defined by

$$Q = \frac{1}{2\zeta} = \frac{\sqrt{Km}}{C} \qquad (7\text{-}4)$$

(4) In figure 7-2, the XZ and YZ planes are planes of symmetry, since the four resilient supporting elements are identical and are located symmetrically about the vertical Z axis. Vertical motion of the foundation, Z_o, $\dot{Z}_o$, or $\ddot{Z}_o$, will result in vertical translation of the resiliently supported mass. This motion can be expressed as follows:

$$m\ddot{Z} + C(\dot{Z} - \dot{Z}_o) + K(Z - Z_o) = 0 \qquad (7\text{-}5)$$

where

m	=	Mass of the resiliently supported body
C	=	Total damping coefficient (C = 4c, where c is the damping coefficient for each spring)
K	=	Total spring constant (K = 4k, where k is the spring constant for each spring)
$Z, \dot{Z}, \ddot{Z}$	=	Displacement, velocity, and acceleration of the mass
$Z_o, \dot{Z}_o$	=	Displacement and velocity of the foundation

(5) Setting relative displacement $Z_r = Z - Z_o$, equation 7-5 is changed to:

$$\ddot{Z}_r + \left(\frac{C}{m}\right)\dot{Z}_r + \left(\frac{K}{m}\right)Z_r = -\ddot{Z}_o(t) \qquad (7\text{-}6)$$

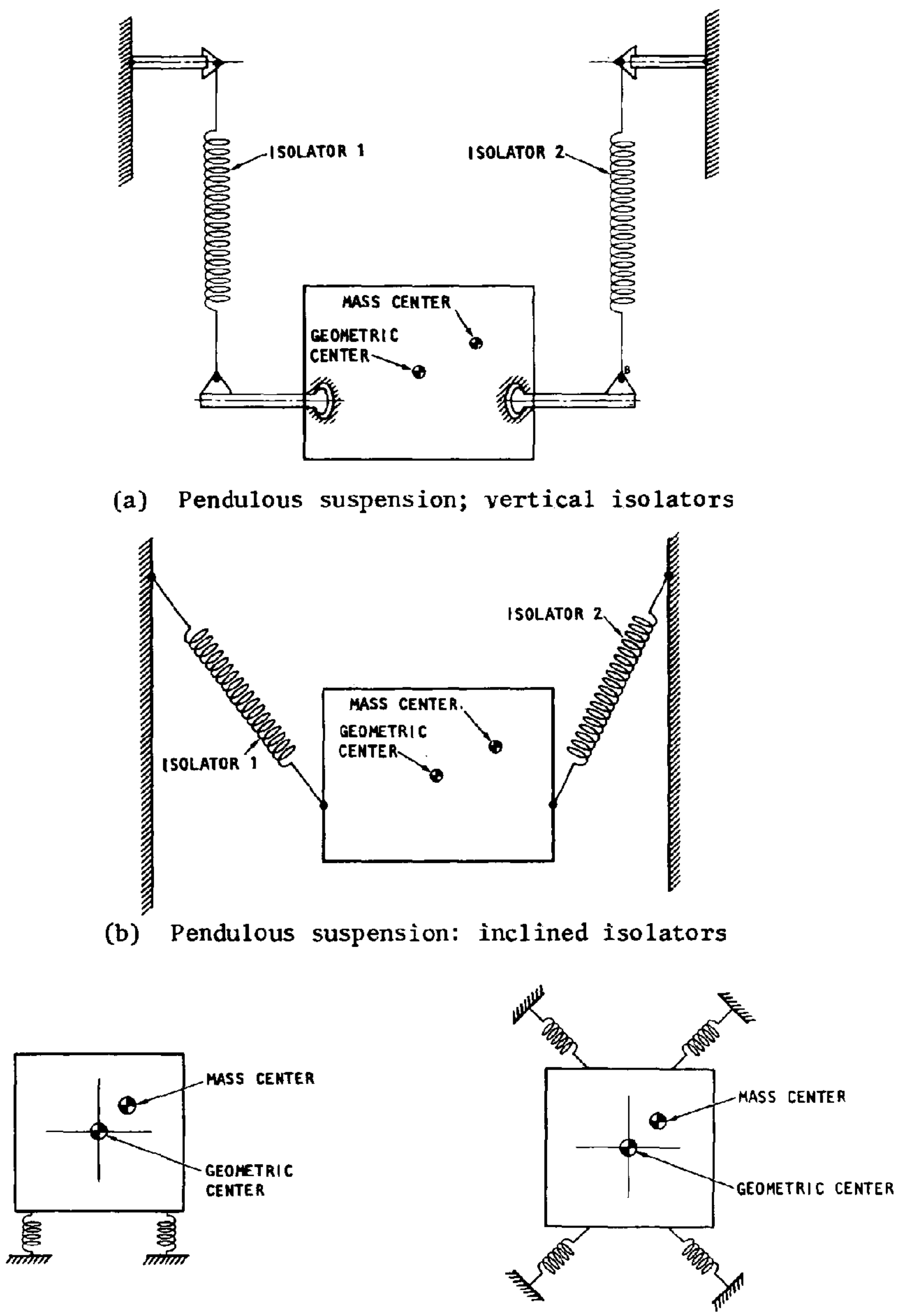
and as $\omega_n^2 = \frac{K}{m}$, $\zeta = \frac{C}{C_c} = \frac{C}{2\sqrt{Km}}$, and $f(t) = -\frac{\ddot{Z}_o}{\omega_n^2}$,

$$\ddot{Z}_r + 2\zeta\omega_n\dot{Z}_r + \omega_n^2 Z_r = \omega_n^2 f(t) \qquad (7\text{-}7)$$

with $\omega_n = \sqrt{\frac{K}{m}}$ being the undamped natural frequency.

(6) If the excitation function f(t) is a simple harmonic function with frequency ω, amplitude a, and phase angle ϕ:

$$f(t) = a\cos(\omega t + \phi) \tag{7-8}$$



(a) Pendulous suspension; vertical isolators

(b) Pendulous suspension: inclined isolators

(c) Base-mounted system

(d) Symmetric suspension

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Figure 7-1. Alternative Isolator Configurations

The solution to equation 7-7 then consists of two parts: a free vibration superimposed on a steady forced vibration (complimentary function and particular solution). The result is

$$Z_r = Be^{(-\zeta+j\sqrt{1-\zeta^2})\omega_n t} + \frac{Ae^{j\omega t}}{\left[1-\left(\frac{\omega}{\omega_n}\right)^2 + j2\zeta\frac{\omega}{\omega_n}\right]} \tag{7-9}$$

where B is a complex constant of integration dependent upon initial conditions (starting transient). When the transient has disappeared, steady-state harmonic motion results at the same frequency as the excitation input. In addition to solutions for sine wave inputs, solutions are also available for excitation inputs of step functions, square waves, triangular pulses, half-sine pulses, etc. (Jacobsen-Ayre, 1958).

(7) Setting $\dot{Z} = j\omega Z$ and $\ddot{Z} = -\omega^2 Z$ and transforming f(t) to the frequency domain as $F(\omega)$, equation 7-7 can be represented as a system function (sometimes called magnification factor) $H(\omega)$:

$$H(\omega) = \frac{\ddot{Z}}{F}(\omega) = \frac{-\omega^2}{\left[1-\left(\frac{\omega}{\omega_n}\right)^2 + j2\zeta\left(\frac{\omega}{\omega_n}\right)\right]} \tag{7-10}$$

$H(\omega)$ may be interpreted as the complex ratio of the acceleration of the mass to an externally applied force. Other forms of this function may also be used, such as the complex ratio of velocity to force:

$$H(\omega) = \frac{\dot{Z}}{F}(\omega) = \frac{j\omega}{\left[1-\left(\frac{\omega}{\omega_n}\right)^2 + j2\zeta\left(\frac{\omega}{\omega_n}\right)\right]} \tag{7-11}$$

and the complex ratio of displacement to force:

$$H(\omega) = \frac{Z}{F}(\omega) = \frac{1}{\left[1-\left(\frac{\omega}{\omega_n}\right)^2 + j2\zeta\left(\frac{\omega}{\omega_n}\right)\right]} \tag{7-12}$$

(8) The real and imaginary parts of H and the magnitude $|H|$ are given in figure 7-3 as functions of ω for moderate damping ($\zeta = 0.1$ or $Q = 5$). When $\omega << \omega_n$, the output is nearly in phase with the excitation; when $\omega >> \omega_n$, the displacement lags the excitation by almost 180 deg; the change in phase is most drastic in the neighborhood of $\omega = \omega_n$, where the phase lag is 90 deg.

(9) When the damping is light ($Q \geqslant 10$) the resonance peak in $|H(\omega)|$ occurs approximately at $\omega = \omega_n$ and the curve is approximately symmetrical for small variations in ω about $\omega = \omega_n$. The peak amplitude is approximately $Q = \frac{1}{2}\zeta$ and the amplitude falls to 0.707 times this at the points P_1 and P_2 with frequencies $\omega_n(1 \pm \frac{1}{2}Q) = \omega_n(1 \pm \zeta)$, as shown in figure 7-4. These points are called the *half-power* points because the power that can be absorbed by a dashpot from a simple harmonic motion at a given frequency is proportional to the square of the amplitude. The frequency difference between the half-power points is often referred to as the *bandwidth* of the system, or $BW = \omega_n/Q = 2\zeta\omega_n$.

(10) Another useful ratio is the complex ratio of the relative displacement (Z_r) to the foundation displacement (Z_o), as given in equation 7-13 and sketched in figure 7-5.

$$\tau(\omega) = \frac{Z_r}{Z_o}(\omega) = \frac{\left(\frac{\omega}{\omega_n}\right)^2}{\left[1-\left(\frac{\omega}{\omega_n}\right)^2 + j2\zeta\left(\frac{\omega}{\omega_n}\right)\right]} \tag{7-13}$$

(11) As discussed in chapter 3, transfer functions are ratios of like variables, such as force-out to force-in or acceleration-out to acceleration-in. The transfer function in the vertical direction for figure 7-2 is expressed for ratios of displacement, velocity, and acceleration of mass and of base motions as:

$$\tau(\omega) = \frac{Z_o}{Z_g} = \frac{\dot{Z}}{\dot{Z}_o} = \frac{\ddot{Z}}{\ddot{Z}_o} = \frac{\left[1 + j2\zeta\left(\frac{\omega}{\omega_n}\right)\right]}{\left[1-\left(\frac{\omega}{\omega_n}\right)^2 + j2\zeta\left(\frac{\omega}{\omega_n}\right)\right]} \tag{7-14}$$

The absolute value of equation 7-14 is the same as the function sketched in figure 7-3.

(12) Response time histories may be determined for the resiliently supported mass from the expressions of equations 7-10, 7-11, and 7-12 by complex multiplication with the transient input force in the frequency domain and then inverse transformation to the time domain:

$$Z(t)\,(=)\,Z(\omega) = F(\omega)\,H(\omega) \tag{7-15}$$

where

- $Z(t), Z(\omega)$ = Time history and frequency response of spring mass
- $F(\omega)$ = Fourier transformation of input forcing function, f(t)
- $H(\omega)$ = System function of shock isolation system

The transient input force f(t) must first be Fourier-transformed to $F(\omega)$ and then complex-multiplied with $H(\omega)$. $Z(\omega)$ is the frequency response of the resiliently supported mass; when $Z(\omega)$ is inverse-transformed, it yields the time history response. Solutions of this type require a digital computer equipped with a fast Fourier algorithm.

(13) Base motions of acceleration, velocity, or displacement time histories are similarly used to compute the response of the isolated mass by using the transfer function of equation 7-14 as:

$$Z(t)\,(=)\,Z(\omega) = Z_o(\omega)\,\tau(\omega) \tag{7-16}$$

where

- $Z(t), Z(\omega)$ = Time history and frequency response of spring mass
- $Z_o(\omega)$ = Fourier transformation of base motion time history, $Z_o(t)$
- $\tau(\omega)$ = Transfer function of shock-isolated system

This method of response computation is also applied to much more complicated systems than that shown in figure 7-2. The method can be applied to both analytic and measured transfer and system functions. The plots of transfer functions shown in figures 5-13 to 5-22 were convolved with facility base motions to predict motion-time histories on the isolated platforms (Safford-Walker, 1975). With the motion time histories on an isolated platform, shock-response spectra can be easily calculated.

(14) Another form of solution to equation 7-6 is by Laplace transformation:

$$\ddot{Z} + 2\zeta\omega_n\dot{Z} + \omega_n^2 Z = -\ddot{Z}_o(t) \tag{7-6}$$

$$(s^2 + 2\zeta\omega_n s + \omega_n^2)z(s) = -\mathcal{L}\ddot{Z}_o(t) = z_o(s)$$

$$z(s) = \underbrace{\frac{1}{\left[(s+\zeta\omega_n)^2 + \omega_n^2(1-\zeta^2)\right]}}_{\text{system function}} \cdot \underbrace{\mathcal{L} - \ddot{Z}_o(t)}_{\text{input function}} \tag{7-17}$$

and by inverse Laplace transform to the time domain:

$$Z(t) = \mathcal{L}^{-1}\,z(s) \tag{7-18}$$

For a wide variety of transient input functions and for not-too-complicated models of shock isolation systems, Laplace solutions can be readily obtained with the use of small calculators and tables of Laplace transforms.

(15) Horizontal base motion in either the x or y axis of figure 7-2 will induce translation and rotation in the shock-isolated system simultaneously. Base motion (X_o) of the system in the x-direction results in the following differential equations of motion:

$$m\ddot{X} + 4C_x(\dot{X} + a_Z\dot{\beta} - \dot{X}_o) + 4k_x(X + a_Z\beta - X_o) = 0$$

$$I_y\ddot{\beta} + 4C_Z a_x^2\dot{\beta} - 4a_Z C_X(\dot{X} + a_Z\dot{\beta} - \dot{X}_o) + 4k_Z a_x^2\beta - 4a_Z k_x(X + a_Z\beta - X_o) = 0 \tag{7-19}$$

Solutions to equation 7-19 are found similarly to that for the previously described vertical motion case. The transfer function magnitude at point 1 of figure 7-2 for equation 7-19 is given in figure 7-6 for specific parameters. Changing parameters will shift both the resonances and amplitudes in the curve.

(16) The foregoing discussion of rigid body modeling of a linear shock isolation system was provided to indicate a method for preliminary analysis. A comprehensive study of rigid-body isolation is provided in chapter 3 (by Himelblau and Rubin) and chapter 40 (by Crede and Ruzicka) of Harris-Crede (1976). These chapters are recommended for additional information of use in actual applications.

c. *Nonlinear systems*

(1) Nonlinear systems yield responses that are other than directly or inversely proportional to a given input variable. Many shock isolation systems contain nonlinear elements either by design or from the physical nature of the isolation elements.

(2) Many spring elements contain nonlinear or stiffening springs near the limits of their maximum displacements to prevent impact bottoming or extension beyond the spring's stress limits. Pendulum systems incorporating a spring in series also have nonlinear characteristics (Parsons, 1962). Damping elements are available with a variety of physical properties other than the classical velocity proportional. These nonlinear damping devices are classed as Coulomb (friction), hysteretic, nonproportional (velocity-squared), discontinuous, and servocontrolled.

(3) Constant-force isolation devices such as the rolling ring torus and versions of the liquid spring (chapter 6) are also classed as nonlinear. Impact of the isolated platform or equipment with the rattlespace boundaries, or the use of snubbers (foam bumpers) to absorb rattlespace impact, are other highly nonlinear effects that must be considered.

(4) The dynamic equations of motion of a nonlinear discrete mass system are formed, and the coupled response is obtained in steps corresponding to specific instants of time. The general solution approach on digital computers is that at a given time the motions are held fixed so that the internal forces can be calculated. Based on the response motion and internal forces at earlier times, the responses at the time in question are predicted. With the predicted values of the motion, the internal forces are recalculated. Using these forces, a

new response motion can then be determined. Therefore, at each time step, the relationship between the response motion and the internal forces is satisfied. This process is continued step-by-step throughout the time interval of interest. The numerical method used for this step-by-step solution involves two types of integration formulas. The fourth-order Adams-Bashforth "2/3" predictor-corrector method (Ralston-Wilf, 1964), which is a stable solution method, is used for most of the solution. However, this method is not self-starting, so the fourth-order Runge-Kutta method is used to start the solution process.

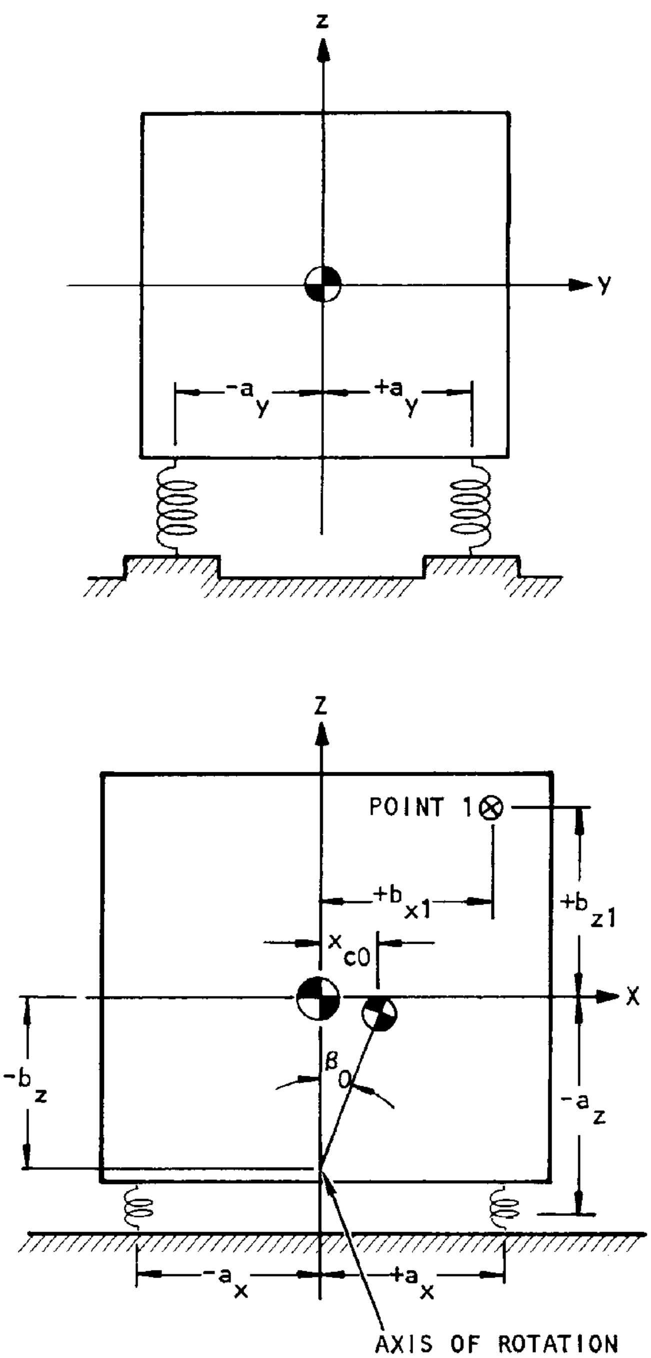

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Figure 7-2. Example of a Rigid Body on Orthogonal Resilient Supporting Elements with Two Planes of Symmetry

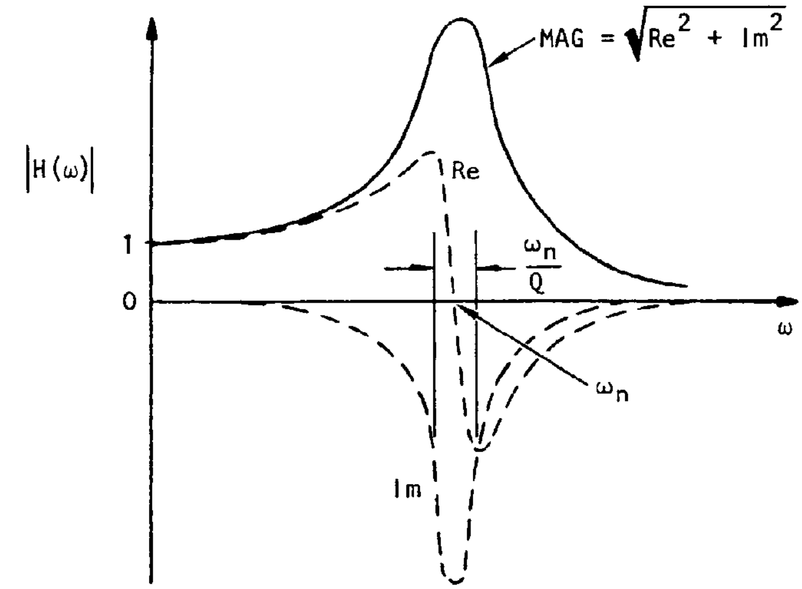

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Figure 7-3. System Function for Single-Degree-of-Freedom System

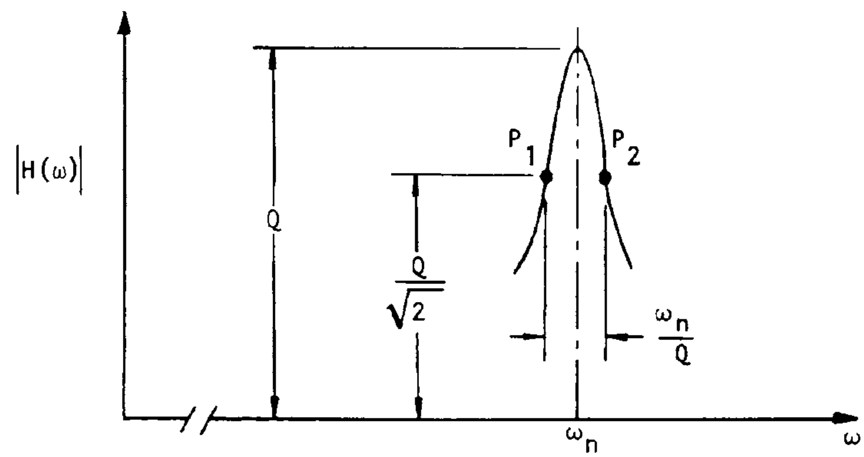

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Figure 7-4. Half Power Points and Bandwidth of Lightly Damped Single-Degree-of-Freedom System

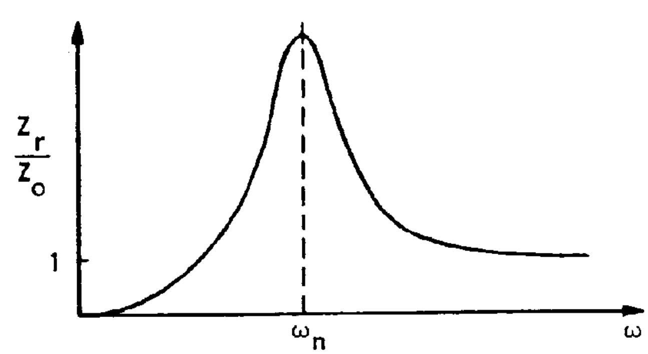

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Figure 7-5. Ratio of Relative Displacement to Displacement of Base

(5) In addition to the step-by-step solution of the nonlinear equations of motion, the natural frequencies and mode shapes of the elastic system can also be obtained by solving the eigenvalue problem using the sweeping technique (Hurty-Rubinstein, 1964). This technique uses a power method (i.e., an iterative method) and the orthogonality relationships among the normal modes.

(6) Sensitivity studies similar to those for the rigid-body linear system should also be run to account for statistical variations and uncertainties. In addition, expected waveform variations of input motions from the facility must be determined due to isolation system nonlinearities. A unique input will explicitly define a unique response; whether or not the response represents maximum notions can be determined only by identification and systematic variation of critical input parameters.

d. Optimization

(1) Optimization is implicit in the design process. Isolation devices act to reduce the undesirable effects of shock and vibration on critical equipment elements and personnel. Selection of isolators for optimum system performance requires the following steps (Pilkey-Pilkey, 1975; Sevin-Pilkey, 1971):

—Define the independent parameters of the problem.

—Mathematically define the constraints, which are the limits imposed on each of the system parameters or any particular combination of them.

—State the functional constraints, which are the physical laws governing the behavior of the system under consideration.

—Develop a suitable criterion for decision that mathematically expresses the objective of the designer. It is possible that this criterion will vary according to the judgment or goals of the individual designer.

—Develop effective search techniques for systematic investigation of feasible solutions for the design with the highest possible merit value.

The first three steps deal with the description of the design domain, which includes all feasible solutions. The boundaries of this domain are defined by the constraints. The last two steps define the objective and the search strategy to attain this objective.

(2) A shock isolation system can be defined as "optimum" if the relative displacement between the base, support, or the facility and the isolated item (the rattlespace) is minimized for a given allowable maximum acceleration experienced by the isolated item, or if the maximum acceleration of the isolated item is minimized for a given amount of rattlespace, for any specific forcing function of interest.

(3) The above may further be illustrated by figure 7-7, where the limiting performance characteristic is

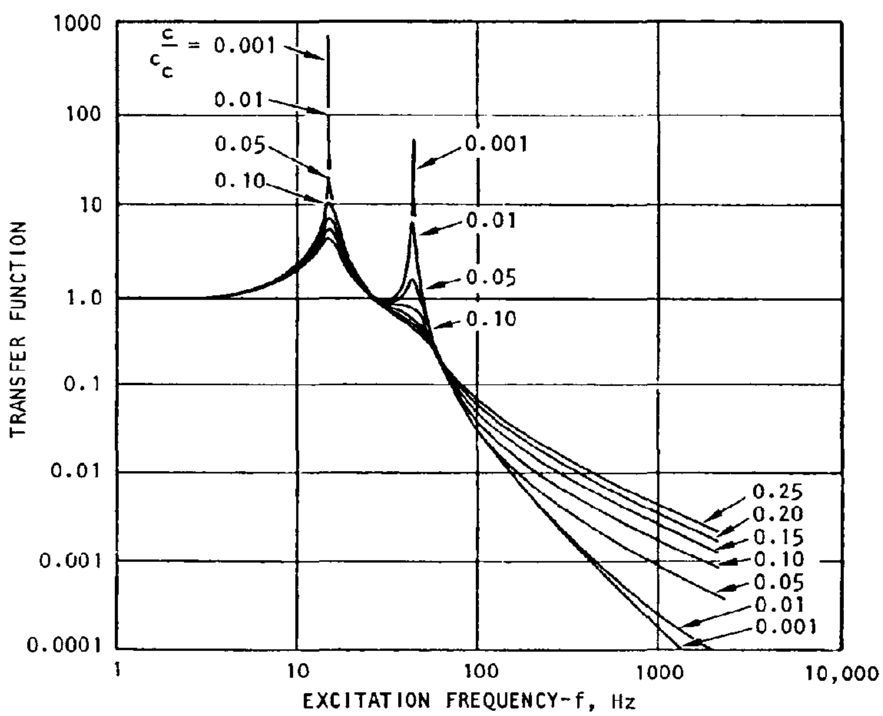

Figure 7-6. Transfer Functions for System Shown in Figure 7-2 at Point 1 for Horizontal Base Motion (X Direction) (Harris-Crede, 1976)

shown in normalized form relative to the maximum displacement D_f and maximum acceleration A_f of the input disturbance. In this form, the intercepts are the points (1, 0) and (0, 1) which correspond to the extreme situation of the rigid connection and a constant-force isolator, respectively. In other words, as the rattlespace constraint approaches zero, the optimum isolator will transmit an acceleration approaching that of the base motion. Alternatively, as the rattlespace constraint approaches the base displacement, the optimum isolator will transmit a vanishingly small acceleration. While isolator performance is of interest between these limits, it may be noted that nonoptimum isolators easily can exceed them. For example, a sufficiently stiff, but not rigid, linear spring will transmit twice the peak base acceleration, whereas the mass could undergo an excursion far in excess of the base displacement if a resonant condition develops (Sevin-Pilkey, 1971).

(4) The studies of Platus (Platus et al., 1973; Platus, 1973) and supported by Liber and Sevin (1966) and Klein (1971) led to the conclusion that for many waveforms, a constant force isolator provides the best possible solution in the tradeoff betwen shock attenuation and rattlespace. Certain problems remain to be explored in constant force isolators with respect to seal friction (deadband or uncertainty in equilibrium position), overshoot and performance reduction when used with mechanical cables, efficiency of liquid springs in approaching constant force characteristics, high frequency transmission, and effect of low velocity impacts.

(5) Optimization studies have evolved over the years (Liber-Sevin, 1966; Klein, 1971; Platus et al., 1973; Platus, 1973) and have led in large part to the development of the computer codes listed in appendix B.

7-3. Multidegree-of-freedom analysis.

a. General. The analysis of multidegree-of-freedom systems can be described more specifically as the analysis of discrete-parameter systems consisting of a finite number of lumped masses interconnected by springs and dashpots. The number of degrees of freedom of a system equals the number of independent coordinates necessary to completely specify the configuration of the system. Figure 7-8 illustrates a typical lumped parameter model of a shock-isolation system.

b. Lumped parameter method.

(1) By applying Newton's laws of motion to each component of the lumped parameter system with n degrees of freedom, a set of n coupled differential equations of motion is obtained. In general, these equations can be expressed in matrix notation as follows:

$$[m]\ddot{\vec{x}} + [C]\dot{\vec{x}} + [K]\vec{x} = \vec{F}(t) \qquad (7\text{-}20)$$

where

$[m]$ = Mass matrix of order n x n
$[C]$ = Damping matrix of order n x n
$[K]$ = Stiffness matrix of order n x n
$\vec{x}$ = Displacement vector of order n
$\dot{\vec{x}}$ = Velocity vector of order n
$\ddot{\vec{x}}$ = Acceleration vector of order n
$\vec{F}(t)$ = Excitation vector of order n

(2) In general, multidegree-of-freedom shock isolation systems give rise to nonlinear differential equations of the type shown in equation 7-20. For example, the presence of snubbers and velocity-dependent

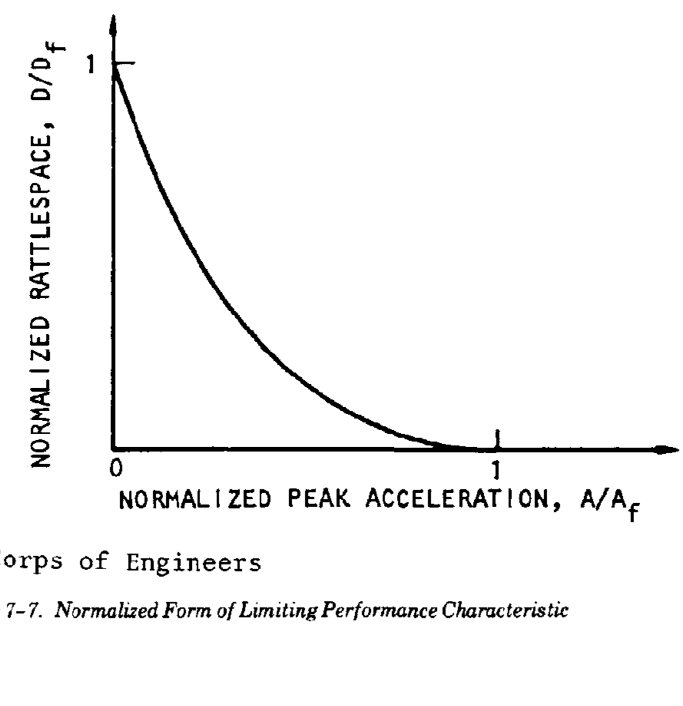

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Figure 7-7. Normalized Form of Limiting Performance Characteristic damping elements results in amplitude-dependent restoring forces. Typical nonlinearities that are encountered in practical cases are shown in figure 7-9.

c. Solution techniques. The nature of the dynamic system has a significant influence on the methods of solution that can be used in conjunction with equation 7-20.

(1) Linear systems. Given a linear elastic structure and a specific excitation time history, then the most commonly used methods to solve its governing equation of motion as expressed in the form of equation 7-20 are as follows:

(a) Solve the equation directly by some finite difference or numerical integration scheme. (For a detailed discussion, see Bathe & Wilson, 1976.) In the direct integration of equation 7-20, a step-by-step numerical integration procedure is used to obtain the solution by the time-marching techniques, in which equation 7-20 is satisfied only at discrete time intervals Δt apart. It is implicitly assumed in this approach that the displacements, velocities, and accelerations within each time interval Δt is "smooth" enough that it can be approximated by an assumed form (e.g., linear variation of acceleration). The commonly used direct integration methods are the central difference method, the Houbolt method, the Wilsom θ method, and the Newmark method.

(b) Find the natural frequencies and mode shapes to decouple the equations, solve the decoupled equations, and then find the response variables of interest. Details regarding this subject can be found in Biggs (1964) and Clough and Penzien (1975). Standard computer routines are available to obtain the eigenvalues (natural frequencies) and eigenvectors (characteristic shapes) of general linear systems. An extremely important property of normal modes is the fact that any two modes are orthogonal. This can be expressed in the form:

$$\sum_{r=1}^{j} m_r \, a_{rn} \, a_{rm} = \delta_{nm} \, m_n \qquad (7\text{-}21)$$

where subscripts

n,m = Any two normal modes of the system

r = Index referring to the *rth* mass out of a total j masses

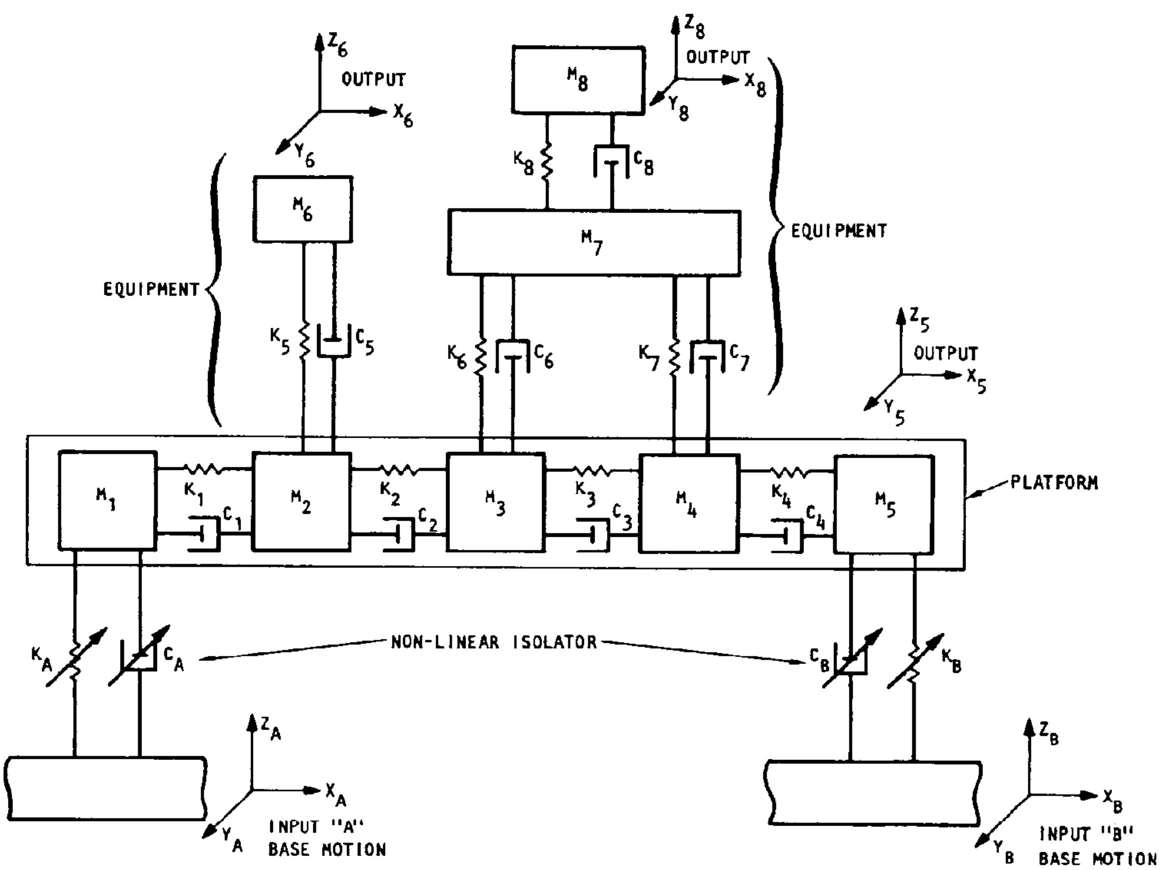

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Figure 7-8. Lumped Parameter Model of a Shock Isolation System

a_{ij} = Magnitude of eigenvector (mode shape) number j at location number i, i.e., $a_{ij} = A_i(j)$

The symbol δ_{nm} represents the Dirac delta function defined as

$$\delta_{ij} = \begin{cases} 0 & i \neq j \\ 1 & i = j \end{cases} \qquad (7\text{-}22)$$

With the help of the orthogonality condition, the solution of equation 7-20 with a slight amount of damping and subjected to base excitation $\ddot{y}(t)$ is

$$R_i = \sum_{r=1}^{n} f(\omega_r) \{P.F.\}_i^{(r)} S_v^{(r)}(t) \qquad (7\text{-}23)$$

where R_i is the response (displacement, acceleration, etc.) of mass m_i

$f(\omega_r)$ = A function of the natural frequencies of the system,

$f(\omega_r) = \dfrac{1}{\omega_r}$ if $R_i = z_i = (x_i - y)$ or $f(\omega_r) = \omega_r$

if $R_i = \ddot{x}_i$

y = Displacement of the base motion

x_i = Absolute displacement of mass m_i

z_i = Relative motion of m_i with respect to its support

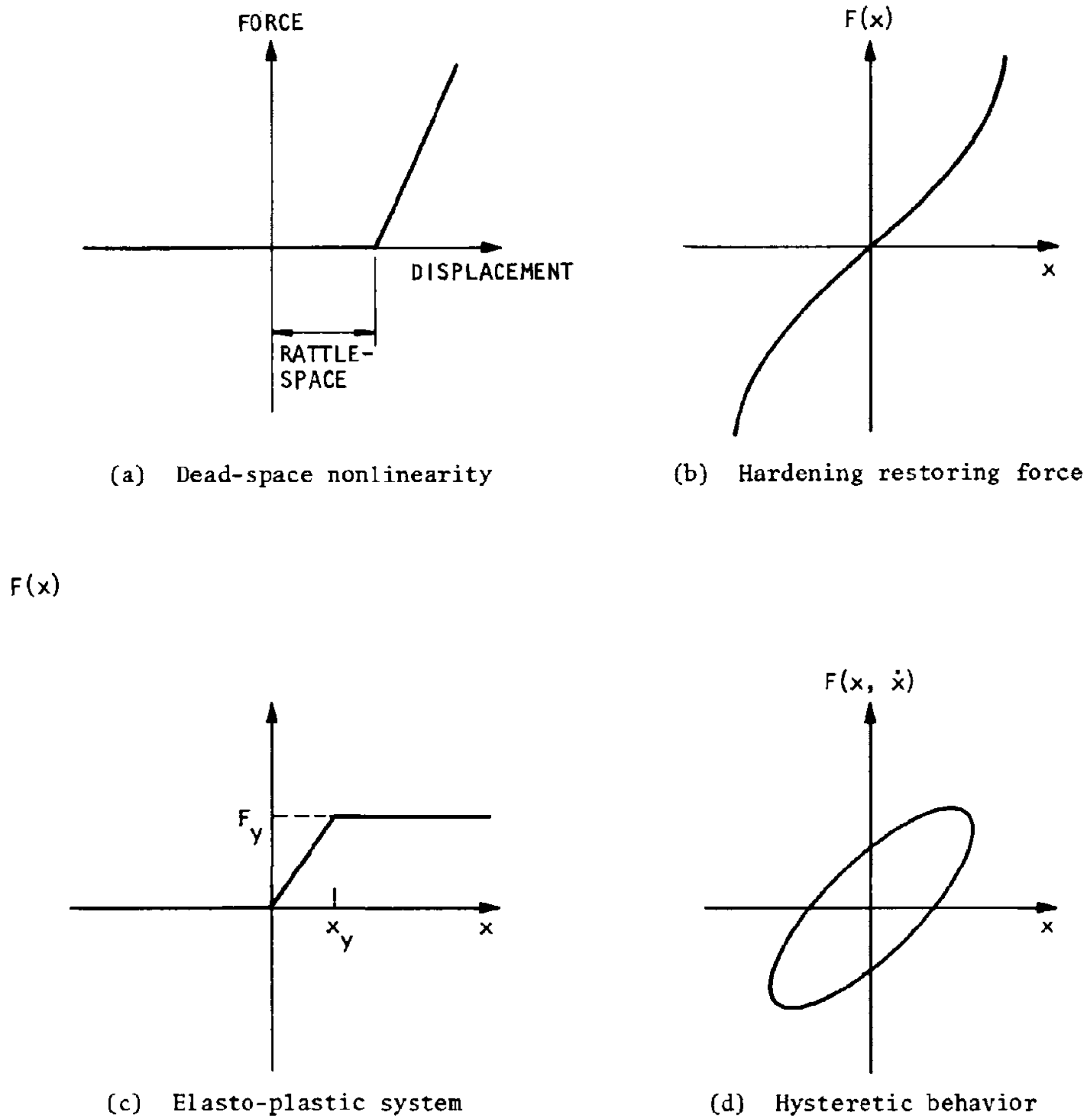

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Figure 7-9. Typical Nonlinearities

$$\{P.F.\}_i^{(r)} = \frac{A_i^{(r)} \sum_{j=1}^{n} m_j A_j^{(r)}}{\sum_{j=1}^{n} m_j A_j^{(r)^2}} \qquad (7\text{-}24)$$

is a modal participation factor

A_i = *ith* mode shape

n = Degrees of freedom of system

$$S_v^{(r)}(t) = \int_0^t \ddot{y}(\tau) e^{-\omega_r \zeta_r (t-\tau)} \sin \omega_r (t-\tau)\, d\tau \qquad (7\text{-}25)$$

$\ddot{y}(t)$ = Support acceleration time history

ω_r = *rth* eigenvalue (frequency) of the system

ζ_r = Ratio of initial damping in mode r of the structure

The general form of R_i, the response parameter from equation 7-23, is

$$R_i = \sum_{r=1}^{n} R_i^{(r)}(t) \qquad (7\text{-}26)$$

Equation 7-26 can be applied as is to obtain the exact solution for modal superposition. For conservative estimates, R_i can be found from

$$R_i \leqslant \sum_{r=1}^{n} |R_i^{(r)}|_{max} \qquad (7\text{-}27)$$

The most commonly used method of approximate modal superposition is the square-root-of-the-sum-of-squares method (RSS):

$$R_i = \sqrt{\sum |\bar{R}_i^{(r)}|^2_{max}} \qquad (7\text{-}28)$$

(c) Take the Fourier (or Laplace) transformation of equation 7-20 to convert from the time domain t to the frequency domain ω, so that equation 7-20 becomes

$$(-\omega^2[M] + j\omega[C] + [K])\{x(\omega)\} + \{F(\omega)\} \qquad (7\text{-}29)$$

$$\text{or } \{x(\omega)\} = [H(\omega)]\{F(\omega)\} \qquad (7\text{-}30)$$

where

$$[H(\omega)] = (-\omega^2[M] + j\omega[C] + [K])^{-1} \qquad (7\text{-}31)$$

H(ω) = System transfer function

$j = \sqrt{-1}$

Having equation 7-29, the response $\{x(t)\}$ may be obtained by taking the inverse Fourier transformation. Details regarding this topic are available in Meirovitch (1967).

(d) Without deriving equation 7-20 mathematically find the response $h_{xF}(t)$ at the response points of interest to unit impulses applied at the load points of interest. Then the response to any timewise inputs F(t) are, by the use of the convolution integral for transients,

$$\{x_i(t)\} = \int_0^t [h_{xF}(t-\tau)]\{F(\tau)\}\, d\tau \qquad (7\text{-}32)$$

The impulse response function [h(t)] is the Fourier transformation of the function [H(ω)] in equation 7-30.

(e) Determine the elements of the matrix [H(ω)] directly by using sinusoidal inputs, since for such inputs and responses, the same [H(ω)] appearing in equation 7-30 is obtained, that is

$$\{\bar{x} e^{j\omega t}\} = [H(\omega)]\{\bar{f} e^{j\omega t}\} \qquad (7\text{-}33)$$

The development and use of the dynamic stiffeners functions in [H(ω)] are discussed in detail in Neubert (9177).

(2) Nonlinear systems. For this class of problems, the only feasible analytical technique available relies on the numerical integration approach discussed earlier. A typical computer code the can handle nonlinear analysis and that is in the public domain is NONSAP. This program is capable of solving dynamic response calculations in problems with material as well as geometric nonlinearities. This program is available from University of California, Berkeley, for use on CDC computers, and from University of Southern California for use on IBM computers.

7-4. Finite element analysis.

a. The finite element method is a powerful numerical procedure for solving the mathematical problems of engineering. Its applications range from the analysis of the structural framework of an aircraft or an automobile to that of a complicated thermal/fluid mechanical system. The particular advantages to using the method for structural dynamic problems are the following:

—Generality of the structure or continuum that can be analyzed

—Relative ease of establishing equations of motion

—Good numerical properties of system motions involved

Some of the widely used finite element codes are SAP IV, V, VI, NONSAP, NASTRAN, and STARDYNE.

b. Numerous technical papers and books are available regarding this subject matter. A detailed bibliography of the available publications is given in Bathe-Wilson (1976).

7-5. Rattlespace requirements.

a. Establishment of rattlespace requirements is critical for space allocation in a protective facility and in sizing of the facility. Rattlespace requirements can best be established in terms of probability and confidence (e.g., 1 percent probability at 90 percent confidence) that the shock-isolation system will impact the facility (AA, 1978). Impact also includes extension or compression of isolators to their physical limits (stops).

b. The conventional procedure to determine rattlespace is by calculation (finite element or lumped parameter) of the relative displacement between the shock-isolated platform and the facility. These calculations apply to the rattlespace perimeter of the platform for the worst-case attack directions.

c. There is a strong tendency in facility design to minimize rattlespace. In many cases, severe reductions in rattlespace allowances lead to significantly higher costs and, because systems are more complex, to reduced confidence in the ability of systems to meet the attenuation requirements. For facilities in high-overpressure regions, where the cost-per-unit volume of structure is high, it is expected that optimum overall design should be minimize rattlespace. The resulting loss in confidence in the isolation system in such cases might be regained by more comprehensive analyses and tests. However, in moderate- or low-overpressure regions it is doubtful that significant cost saving can be realized by severely limiting the available rattlespace. Regardless of conditions, the isolation-system designer should make a strong effort to maintain rattlespace requirements within reasonable limits. This can be achieved by careful arrangement of supported equipment and by proper selection of suspension configurations and components. In many circumstances and within certain limits, a reduction in the displacement relative to the facility can be accompanied by a reduction in the severity of the shock-isolated environment. If possible, reductions in rattlespace should be achieved by improvements in efficiency rather than by additional complexity. One method is to group equipment together and to shock mount the complete assembly, thus requiring a much smaller total rattlespace volume than had each of the items been shock-isolated individually. Other methods for minimizing rattlespace are the selection of suspension configurations such that coupling between rigid body modes is minimized, and by separating its modal frequencies.

d. Damping in both rotational and translational modes can also be effective in reducing rotational displacements. As the energy in rotation is due principally to a transfer from a coupled transitional mode, a rapid dissipation of energy in the translational mode will reduce the amount of energy transferred.

e. A preliminary estimate of rattlespace requirements must often be made prior to the selection of the isolation system configuration and components. A rule of thumb is to specify rattlespace in each of the three orthogonal gravity-oriented coordinates equal to approximately twice the displacement indicated on the appropriate actual system in that coordinate. This rule of thumb should be applied only to highly stable systems in which the load is reasonably compact, there is no rotational component of the shock, and the system is well balanced statically and dynamically and is damped. The conservatism of this rule of thumb has been demonstrated both by analysis and by experiment.

f. Uncertainties and random variation applicable to the weapons threat, facility response motion, and isolated platform response motion must be included in the determination of rattlespace. Uncertainties in platform motion prediction may be particularly acute where weights and mass distribution of shock-isolated equipment are not well defined.

g. An alternate procedure for rattlespace determination is in process of implementation under the sponsorship of the Shock Physics Directorate, Strategic Structures Division (SPSS) of the Defense Nuclear Agency (DNA). This rattlespace procedure is a statistical approach where the rattlespace provided is identified as the capacity, C, and the dynamic displacement of the shock-isolated platform or equipment as the demand, D. Then D may be put in the form

$$D = d\,Q \tag{7-34}$$

where

d = Facility shock displacement

Q = Modal coupling factor

Because the shock-isolated platform/equipment is a low-frequency system, the relative displacement between it and the cavity is equal to the absolute displacement of the facility, i.e., the shock isolation system attachment points.

(1) The displacement d represents the displacement of the mass center of the isolated system. The displacement of some corner of the isolated system may exceed the displacement of its mass center because of pitching motion resulting from coupling among the mormal modes of the shock-isolated platform. Such a magnification of the displacement is accounted for by modal coupling factor Q.

(2) To varying degrees, each variable of the Capacity and the Demand embodies both random and nonrandom uncertainty. The random uncertainty, or variability, cannot effectively be reduced by gathering more data or by conducting research and development. The nonrandom uncertainty, on the other hand, can be reduced by gathering more data or by conducting research and development, since it reflects parameter-estimation and modeling errors, i.e., it reflects ignorance. Random uncertainty is used as the basis for probability calculation; nonrandom uncertainty is used as a confidence statement.

(3) Except for separation of random and nonrandom uncertainty for the purposes stated above, the methodology is identical to that of Ang-Cornell (1974). The methodology was published by DNA circa 1979 and may be used for rattlespace determination.

CHAPTER 8

DESIGN PROCEDURES

8-1. Introduction.

a. This chapter describes design procedures for shock isolation systems. Because there are many possible design approaches and analytical sequences in the resolution of a design problem, detailed and fixed step-by-step procedures impose unnecessary restrictions on the solutions and confound one's understanding of the critical design problems. Thus, this chapter presents a sequence of procedures to insure that most of the relevant design alternatives and problems are considered and that the appropriate methods are applied to the design. Lessons learned, conclusions and recommendations regarding the many platforms installed in the Safeguard Missile complex are provided at the end of this chapter in paragraph 8-4 (Bradshaw-Sonnenberg, 1978).

b. The analyses of facility environments, equipment fragilities, and personnel tolerances described in earlier chapters will have indicated the extent to which shock isolation is required or whether hard mounting of equipment is feasible. Tradeoff and cost-effectiveness studies, also alluded to earlier, may have influenced this decision. Accordingly, this chapter discusses two subsequent considerations: The hard mounting of equipment that is sufficiently rugged to withstand the shock-induced loads without failure or malfunction; and design approaches to shock isolation system.

8-2. Hard-mounted systems.

a. When off-the-shelf equipment is available to perform a desired function, it is usually less expensive than specially designed equipment. However, there may not be adequate data on the performance of off-the-shelf equipment under the shock environment. In this case, special proof-testing of such equipment will be required (app. A and chap. 4). In contrast, specially designed equipment, which may cost more to procure or develop, often has more complete performance data and therefore may not require additional testing.

b. Even if tests show that the equipment itself will perform well during and after the shock loading, improperly designed attachments could lead to reduced effectiveness or failure. Therefore, the attachments of the equipment to the structure must be carefully designed to provide support without inducing adverse loads and to avoid permanent displacements, particularly when the position and orientation of the equipment is critical to its function.

c. Once an appropriate test program has established that the equipment can survive the specified shock loadings, the attachment to the supporting structure can be designed. Descriptions of two different approaches for defining attachment design loads follow.

(1) Equipment mounted on nearly rigid structures. When the equipment is attached directly to the external walls or floors of the structure and when the structure is nearly rigid (fig. 8-1a) the following approximate procedure should be used to define attachment design loads:

Step 1: Assume the response acceleration of the equipment at its c.g. is equal to the resultant of the horizontal and vertical accelerations given by the constant acceleration region of the shock spectrum envelope. These accelerations applied to the equipment mass are the forces used for design of the supports.

Step 2: Consider the forces determined in Step 1 to be either in phase or out of phase, and determine the case that causes maximum reactions at the support points. For example, in the symmetrical system shown in Figure 8-1b, the maximum reaction is as follows:

$$F_{1z}\,(\text{tension}) = \frac{F_z}{2} + \frac{F_x h}{2b} \qquad (8\text{-}1)$$
$$F_{1x} = \frac{F_x}{2}$$

The terms in the preceding equation are defined in figure 8-1b.

Step 3: Design the anchors to remain elastic (up to yield) for the combined reactions (worst case) determined in Step 2. If yielding of the anchors or attachment pieces is acceptable (that is, if a ductility ratio of $\mu > 1$ is allowed for the attachment elements), the design may be modified accordingly, based on elastic-plastic shock spectra.

The reactions are based on the modified forces:

$$F_z = M\ddot{z}\,\frac{\mu}{2\mu - 1}\;;\; F_x = M\ddot{x}\,\frac{\mu}{2\mu - 1} \qquad (8\text{-}2)$$

where M is the mass of the supported equipment.

(2) Equipment mounted on flexible structure elements. When the structural elements, or system of elements, on which the equipment is mounted has a natural frequency less than five times the equipment frequency, the flexibility of the system of elements must

be considered when defining the equipment anchor loads. The following approximate procedure should be used in this case:

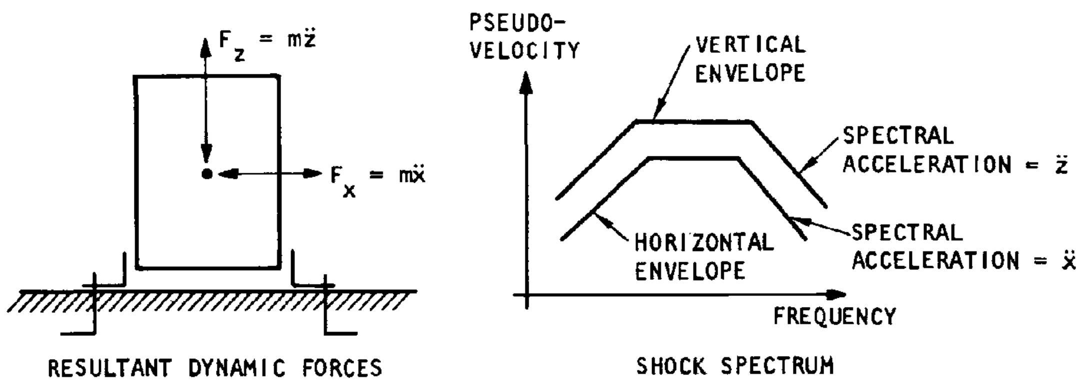
Step 1: Determine the mode shapes, frequencies, and participation factors of the significant modes for the coupled support-structure/equipment system. These parameters can be determined by various methods, depending on the degree of complexity of the structure. For a simple support-structure system, such as the single beam anchored to nearly rigid walls of the hardened facility, approximate methods can be used. If the support-structure complexity does not permit use of approximate methods, then computerized analyses of finite element models of the coupled structure/equipment system should be used. A case of this type might be a multistory support structure within a hardened structure enclosure.

Step 2: Define the damping ratio for all significant modes of the coupled support-structure/equipment system. Approximate techniques for carrying out this step are described in chapter 7.

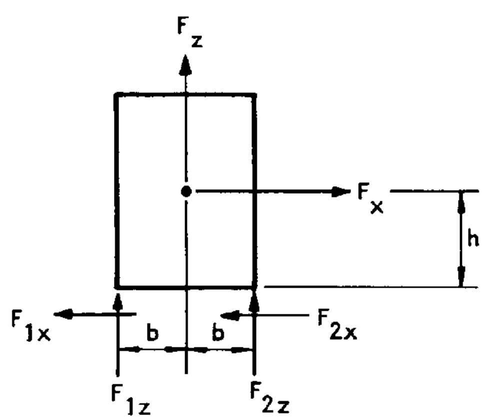
Step 3: Assuming that the input motions at the base of the support structure are defined in terms of response spectra of horizontal and vertical motion, enter each spectrum at the frequencies of the significant modes of vibration to obtain the modal displacement amplitudes.

Step 4: Obtain for each mode of vibration the moments and shear forces at the anchor attachment points of the equipment. Such values, determined analytically, must be obtained once for horizontal input motions and once for vertical input motions. Combine these modal responses using a root-sum-square approach to obtain the total moments and shear forces for each type of input motion.

Step 5: Combine the anchor moments and shear forces from the horizontal input motions with those from the vertical input motions using a root-sum-



(a) Definition of dynamic forces



(b) Determination of reactions for symmetrical system (see Eq. 8-1)

U.S. Army Corps of Engineers

Figure 8-1. Equipment Mounted on to Nearly Rigid Structure square approach. Design the anchors to resist these moments and shear forces.

8-3. Shock-isolated systems.

a. General. When the decision is made to shock-isolate rather than hard-mount the equipment, the next step in the design process is to define as explicitly as possible the performance requirements of the system. The greater the detail of specification, the greater will be the assurance that the system will accomplish its objectives.

(1) Design the shock isolation system to the following functional requirements:

—Reduce input motions to acceptable levels.
—Minimize rattlespace requirements consistent with system effectiveness and cost.
—Minimize coupling of horizontal and vertical motions.
—Accommodate a spectrum of inputs of uncertain waveforms.
—Limit the number of cycles of motion of the isolated body.
—Support the system under normal operating conditions without objectionable motions.
—Maintain constant attitude under normal operating conditions.
—Accommodate changes in load and load distribution.
—Maintain system vibration characteristics over long periods of time.
—Interface properly with other components or parts of the facility.
—Minimize maintenance requirements.

(2) If the shock isolation system is to perform satisfactorily, careful attention must be given to its mechanical design details. Attachments to hard structures must be strong enough to survive the largest acceleration; impact between elements or noisy frictional interfaces must be avoided; installation and maintenance should be made as simple as practical; direct transmission paths for high-frequency motions must be minimized; and elements should have natural frequencies high enough to avoid coupling with the shock isolation system frequencies.

(3) Shock isolation systems are usually designed before the physical characteristics of the equipment to be protected are definitely determined and before equipment dimensions and weights are established. Consequently, the c.g. of the actual system mass seldom lies precisely where it was assumed to be for design. This situation can be somewhat compensated for by adding ballast to the system. Effective ballasting, however, is difficult to achieve in the vertical direction. Even if the mass c.g. is accurately forecast, there remains another condition for dynamic symmetry: that the actual spring stiffness of the isolators is that stiffness assumed in the design. All hardware is manufactured only to some plus or minus tolerance. Accordingly, some imbalance due to variation in isolator and damper properties can be expected. Other practical imperfections that can contribute to imbalance include friction and slight misalignment. Therefore, while the idealized cases and simplified procedures presented herein are adequate for preliminary design and analyses, more sophistication and refinement will be required for a final hardware and system design.

(4) In general, the following major steps may be used as a sequence of design analysis procedures (Saffell, 1971):

—Establish locations, physical properties, and fragility levels of the shock-isolated equipment.
—Select and group equipment to determine the size of platform.
—Define shock environment at the locations.
—Establish performance requirements of the shock-isolation system.
—Select type of shock-isolation system.
—Perform dynamic analysis of selected shock isolation system based on design requirements.
—Establish performance requirements of the shock isolation system components.
—Select type of shock isolation system components.
—Design and analyze shock isolation system components.
—Reevaluate performance requirements based on preliminary design analysis of shock isolation system and components.
—Prepare specifications.

As noted, preparing the specifications is the last step in the design analysis procedures. Shock isolation systems are designed to a set of requirements that are applicable to the specific facilities. However, specifications are written for shock isolation system components that are designed to meet performance requirements of specific shock isolation systems. The specification provides for uniformity of design analysis and for detailed design requirements of system components. Thus, in some cases, specifications may be the final product of a shock isolation system design-analysis effort. The requirements in a specification should include the following minimum items:

—Detailed performance requirement
—Physical size envelopes
—Tolerances
—Requirements for interfacing with other components
—Methods to verify the performance

Finally, the design-analysis procedures are organized so the critical design problems of a shock isolation system are examined in a proper sequence and the necessary analyses are performed without unacceptable assumptions (Saffell, 1971). The final specification must be reviewed for practicality and insurance that the specified requirements are realistic.

(5) As part of the design effort, the shock isolation system designer makes basic design selections with regard to group-mounted or individually mounted equipment, layout of the isolator system, and shock isolation system components. These basic design features will now be discussed.

b. Layout of the isolator system. The large variety of shock-isolation systems can be grouped into four main categories: symmetric, near-symmetric, pendulous, and base-mounted. The characteristics of these categories are summarized in table 8-1. In addition to performance requirements, the selection of a particular type of shock isolation system should be influenced by the following criteria:

—Accessibility of equipment
—Available space
—Input shock environment
—System size and weight
—Location of system c.g.
—Installation
—Maintainability
—Reliability
—Cost

No one type of system will satisfy the requirements of all locations and all equipment installations. Careful consideration should be given to the design criteria and the several shock isolation alternatives before arriving at a final decision. The four types of isolator systems are discussed briefly below:

(1) Symmetric systems. Symmetric systems have isolators arranged so that the system's center of rigidity coincides with its center of mass. The isolators are oriented either parallel to or inclined, relative to the principal axes of the system. Symmetric systems are desirable because of the decoupling of normal modes and the minimizing of transmitted acceleration and velocity.

(2) Near-symmetric systems. Near-symmetric systems consist of isolators arranged so that only a small eccentricity exists between the resultant of the isolator spring forces and the center of mass of the equipment. Small eccentricities often occur because the actual weight and center of mass of the equipment may differ from those assumed in the initial design phase and used to establish isolator locations. Therefore, the final design of the shock-isolated system should account for such eccentricity effects through parametric analyses in which system weight and center-of-mass location are varied.

(3) Pendulous systems. Pendulous systems constitute a special class of symmetric or near-symmetric system. They are used to suspend large bodies from isolators attached to their sides with arms that extend to enclosure support points located above the system c.g. For a bilevel suspension system, isolator spring characteristics and support point locations can be chosen so that there is maximum stability and minimum coupling between modes during dynamic response. Single-level suspension systems are less desirable because of potential stability problems. Areas requiring special consideration for pendulous systems are hardware space requirements, effects of shifts in the center of mass, and damping requirements in the pendulous modes.

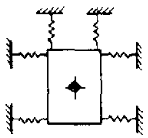
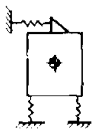
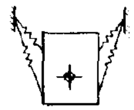
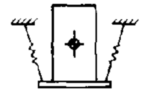
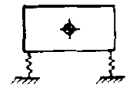
(4) Base-mounted (asymmetric) systems. In base-mounted systems, equipment is supported by vertical and possibly horizontal isolators only along the base of the equipment. A disadvantage of base-mounted systems is their sensitivity to possible variations in the center-of-mass location, isolator spring stiffness, and other system parameters. Also, there are significant dynamic coupling effects. However, such systems may be desirable for any of the following reasons: (a) limited space available (e.g., if the equipment is in a buried protective structure); (b) extensive costs of developing symmetry; and (c) convenience, e.g., many off-the-shelf items of equipment have mounting points already established along their bases.

c. Isolation system components. A shock isolation system has three major components: the platform, the isolators, and the isolator attachments. Since there are many possible component designs, a set of design requirements must be established so the designer can make the proper choice to meet a set of specific performance requirements. Figure 8-2 summarizes major considerations in the design of shock isolation system components. These components will now be discussed in more detail.

(1) Platform design. The shock isolation platform is the structure that supports the isolated equipment and to which the isolators are attached. The basic performance requirement of the platform is that while supporting the isolated equipment, it attenuates motions from the isolators to the platform. To accomplish this, the platform must have specific, bounded stiffness and damping characteristics while staying within the usual engineering design limitations of size, weight, cost, and fabricability.

(a) Design requirements. The major design consideration is the platform stiffness requirement. In association with the equipment and platform weight, the platform stiffness governs the natural frequencies of the assembly. It is desirable to have a rigid platform to avoid couplings of the platform's flexible body motions and the shock isolation system's responses. Usually, a platform may be considered rigid if its lowest flexible-

TABLE 8-1. CHARACTERISTICS OF EXAMPLE SHOCK ISOLATION SYSTEMS (Crawford et al., 1974)

Type	(a) Symmetric		(b) Near-symmetric	(c) Pendulous			(d) Base-mounted
Schematic	 Horizontal and vertical springs-pinned ends	 Inclined Isolator		 Inclined Pendulum	 Pendulum - Low c.g.	 Pendulum - High c.g.	
Feasibility Range	For equipment isolation at low frequency and large relative displacements	For equipment, isolation at mid-range frequency, and small relative displacements	For equipment isolation at low and midrange frequency and low and moderate relative displacements	For total structure isolation at low frequency and large relative displacements	For equipment and personnel platforms at low frequency and large relative displacements	For missile mounts at low frequency and moderate relative displacements	For equipment and total structure isolation at low and midrange frequency and small to moderate relative displacements
Dynamic Coupling	Negligible	Negligible	Small pitch response induced	System characteristics can be selected to minimize coupling	Significant pitch response induced	Very significant	Very significant for acceleration and displacement response
Rattlespace Requirements	Shock spectra values (large horizontal space required for hardware)	Shock spectra values	Shock spectra values are amplified due to pitch response	Shock spectra values are acceptable when the geometry optimized for minimum coupling	Shock spectra are amplified due to pitch response	Shock spectra values are amplified due to pitch response	Shock spectra values are amplified due to pitch response (minimum space required for hardware
Dynamic Stability	Not critical	Not critical	Not critical	Not critical when optimized	May be critical	Very critical	Not critical
Static Stability	Not critical	Not critical (but spring stability must be considered)	Not critical (but spring stability must be considered)	Not critical when optimized	May be critical	Very critical	May be critical (spring stability must be considered)
Nonlinear Effect	Very significant for acceleration response and rattlespace	Not critical	Not critical	Some effect on acceleration response	May affect acceleration response and rattlespace	Significant effect on acceleration response and rattlespace	Not critical

⌖ Location of c.g.

body frequency is four to five times higher than the highest rigid-body frequency of the shock isolation system. Hence, the requirement is to design a platform rigid enough to meet the frequency criteria without paying a penalty for cost. Cost is determined essentially by platform weight for similar fabrication techniques and design. Therefore, it is of paramount importance to distribute the platform structural elements to maximize area moments of inertia while still retaining ease of fabrication and using commonly available structural elements.

1. There are two choices of platform assembly methods. Welding is probably cheaper and results in a more rigid structure. However, the damping inherent in bolted connections may be advantageous for vibration suppression. An interesting approach toward attenuating transmitted high-frequency vibrations would be to damp them out in the platform structure itself by coating the platform with energy absorbent materials. This is commonly done in the automotive industry (undercoating, for example) with excellent results. Several approaches are possible: sprayed or troweled-on mastic; glued-on, asphalt-impregnated felts; and viscoelastic materials. Damping ratios of 20 percent of critical and attenuations of approximately 400 db/sec are common (Harris-Crede, 1976). Manufacturer's literature is quite specific and should be closely followed. The outstanding advantages of using this approach are low costs for material and application, no maintenance, and good damping without introducing additional system resonances.

2. Fabricability is important because it strongly influences cost. Since field fabrication is invariably more expensive and error-prone than shop fabrication, the platforms will normally be shop fabricated and field assembled. Also, because the building structures are often completed prior to installation of shock isolation equipment, size limitations of shop-fabricated platform subassemblies can become critical.

(b) Design concepts. From the basic design requirements, several design concepts can be considered:

- I-beam frame structure
  - Cross braced
  - Diagonal truss
  - Top and bottom cover plates
- Plate box-frame structure
- Tubular space truss structure
- Honeycomb structure

1. For the platform material, structural steel normally is used. Other materials such as steel or wood have been considered. Wood has been experimented with due to its higher damping properties; aluminum has been considered due to its lower weight. For a given platform design, frequencies are about the same should aluminum be considered in place of steel; also an aluminum structure has less total weight. However, the platform frequencies are also influenced by the weight of the mounted equipment. Thus, if the equipment is quite heavy, steel construction should be used because it provides a stiffer platform. Since the platform is designed for stiffness rather than strength, there is no advantage to using high-strength steel.

2. The following two concepts satisfy not only the requirements of rigidity, ease of fabrication, and low cost, they also provide proper attachments for the supported equipment. The shock-isolated platforms may be divided roughly into two groups, depending on the arrangement of supported equipment. The first group supports a few items of heavy and rigid equipment such as motor-generator sets or transformers. The second group consists of many pieces of relatively light and flexible equipment such as control consoles and electrical switch cabinets.

3. The first platform concept is for the first group of heavy equipment. As shown in figure 8-3, it consists of two heavy, wide-flange beams connected by secondary cross beams. The beams are spaced to provide direct supports under the equipment attachment points. Diagonal beams are added to give the necessary torsional stiffness. A thin plate may be installed on top of the structure for servicing and maintaining the equipment. It does not contribute substantially to the stiffness of the platform.

4. The first concept can also be used for the second group of equipment. But, as more and more local stiffeners are needed to properly support the large number of equipments, it becomes no longer cost effective to use heavy beams and diagonal stiffeners. Therefore, the second concept, as shown in figure 8-4, uses a series of box structures made of relatively thin plates. Stiffeners are used at all the equipment attachment locations, and top and bottom plates are necessary to fulfill the bending and torsion stiffness requirements.

5. In the design of a platform, two problems are usually neglected. The first problem is the effect of equipment dynamics on the platform frequencies. Unfortunately, information on the equipment dynamics is usually not available, making heavy reliance on engineering judgment necessary. This is particularly apparent at the modeling of platform and equipment assembly for frequency analysis. The second problem is the proper design of stiffeners for equipment supports. The local equipment-support frequencies should always be checked to satisfy the overall platform stiffness requirements. In this sense, the equipment should never be supported on a thin top plate only.

(c) Design methods. Several computerized methods are available for analyzing platform frequencies (app. B). However, prior to getting detailed results from a digital computer simulation of the platform, preliminary design of the platform must be completed and approximate platform frequencies established. If this is not done, the designer will have no feeling for the validity of the computer solution; and because of a few poorly chosen members, the computer solution may not be valid.

1. To facilitate the preliminary analysis of the platform frequencies, analytical methods for calculating the first-mode torsional and bending frequency will now be discussed. For these approximations to yield a reasonably close solution, the following assumptions are made:

—The platform structure is of steel and has uniform bending torsional stiffness along its length.

—The equipment mass distribution is close to symmetrical but not necessarily uniform from platform midpoint.

Since the applicable platform frequencies are functions only of inertia and stiffness, these properties must first be calculated. The moment of inertia may be calculated from any standard reference on structure design. If the platform cross section can be approximated by a box shape, as shown in figure 8-5, then the approximate bending moment of inertia about axis x-x can be determined as well as the torsional stiffness about the z-z axis.

2. Assuming a platform and equipment arrangement as shown in figure 8-6, the bending frequency in Hz is

$$f_B = \frac{10^6}{2\pi}\sqrt{\frac{7I_{xx}}{WL^3}} \tag{8-3}$$

where I_{xx} = bending moment of inertia, W is the total weight of platform and equipment, and the torsional frequency in Hz is

$$f_T = \frac{10^3}{2\pi}\sqrt{\frac{24J_{yy}}{LI}} \tag{8-4}$$

where I is the mass moment of inertia of the platform and supported equipment about the axis yy and J_{yy} is the torsional stiffness about the axis yy.

3. If the equipment weight is evenly distributed along the length of the platform, the frequency will be about three times higher than the calculated values. The above approximations do not include the stiffness or the frequency of the equipment. If the equipment is significantly stiffer than the platform, then the calculated results are conservative. On the other hand, interactions of equipment and platform dynamics can be evaluated only by more elaborate methods.

(2) Isolators. The isolators supporting the shock isolation platform must satisfy both functional and economic requirements. The functional requirements are determined by the performance requirements of the shock isolation system and the findings from the dynamic analysis of the system. The isolator's ability to meet these requirements depends on its stiffness and damping characteristics, as well as its predictability and reliability of performance. Isolators are usually classified by their load-bearing mechanism, the elastic element. Since the damper element can be added separately, it is usually not peculiar to a specific elastic element design. Isolator elastic elements may be broadly grouped into four categories: Mechanical springs, liquid springs, pneumatic springs, and elastomers. Table 8-2 summarizes the performance, cost, and maintainability characteristics of these four types of isolators. The individual isolators are discussed in detail in chapter 6. The following supplementary information will help the shock isolation system designer select and size these elements.

(a) Mechanical springs. Under this heading fall the common helical and torsional springs and the less common volute and Belleville springs. All of these springs in their nonprecision forms are easily designed and manufactured and are therefore inexpensive. Their value to isolator designers lies in their low procurement cost, design simplicity, and low maintenance requirements. Unfortunately, the cost of mechanical springs rises rapidly when precision and quality control requirements become stringent.

1. Mechanical springs transmit energy through deformation of the structural material of the springs. These deformations can be in axial compression or tension (helical coil springs or rubber shock mounts); bending (Belleville springs or flat springs); or torsion (torsion bars). The springs may be made of metal or elastomers.

2. The accurate prediction of isolator stiffness is paramount for uncoupled platform motion, but mechanical spring rate tolerances of 10 percent are common. The spring rate of a mechanical device usually cannot be adjusted. Belleville springs and variable diameter and pitch springs are exceptions. The correct specification for spring-stiffness values presupposes a knowledge of suspended load, since their ratio must be close to identical for all isolators, for minimum uncoupled motion. Unfortunately, specification of spring loads is also subject to gross errors because mass distribution of equipment on the platform is not accurately known. Other difficulties associated with mechanical springs are hard noise paths, surging, high-frequency transients, and almost insignificant inherent damping. Since damping is required for many shock isolation systems, a separate damping mechanism will be necessary for such systems. This can significantly increase the isolator cost, complexity, and maintenance requirements.

3. In summary, if performance accuracy and damping requirements are not stringent, the mechanical spring is very attractive. For more precision applications or those requiring large damping, the mechanical spring may not even be competitive. The different types of mechanical springs are discussed below.

(b) Helical coil springs. The usual specifications for a spring will include at least the maximum load and either the spring rate or an allowable deflection under this load. When used as isolators in protective construction, a dynamic deflection should also be specified. With this guidance, it is possible to determine the required spring characteristics. For a given maximum static load and allowable deflection under this load, the axial spring rate is given by

$$k = \frac{W}{\Delta_{st}} \tag{8-5}$$

where
W = Axial load
Δ_{st} = Static deflection

The axial spring rate of a helical coil spring is

$$k = \frac{d^4 G}{8D^3 N} \tag{8-6}$$

where
d = Spring bar diameter
D = Nominal spring diameter
G = Shear modulus for spring material
N = number of active coils

1. A helical coil spring is made up of two classes of coils, depending on their function in the spring. The active coils are those coils responsible for the spring's response to loads or displacements. The inactive coils are those coils at the ends of the spring that help provide attachment. The number of active coils depends on the required spring characteristics. The number of inactive coils depends on the treatment of the spring ends. In most shock isolation applications, the end will typically be squared and ground. The number of inactive coils for this condition is two, one at each end.

2. From equation 8-6 it is seen that a large number of combinations of bar diameter, spring diameter, and number of active spring coils will satisfy a given value of spring rate. There are other criteria, however, that must be satisfied, and these criteria help narrow the number of workable combinations.

3. The spring will normally be required to deflect some specified amount without bottoming out. The bottoming-out condition occurs when the spring is fully compressed with no space remaining between coils. The total deflection that must be accommodated is given by

$$\Delta_t = C_f (\Delta_{st} + \Delta_{dyn}) \tag{8-7}$$

where
C_f = Factor of safety against bottoming
Δ_{st} = Static deflection of spring
Δ_{dyn} = Dynamic deflection of spring

4. Although permanent set may be acceptable in some instances, it is normally required that the system return to its original position. This can be accomplished in various ways, but the most common approach in the case of helical coil springs is to prevent inelastic action. In order to prevent inelastic deformation of the spring in the event of overload, coil springs are usually designed so that the elastic shear strength of the spring is not exceeded when the spring is fully compressed. The load corresponding to full compression is given by

$$p^* = k\,\Delta_t \tag{8-8}$$

and the maximum shear stress at full compression is given by

$$\tau_{max} = K\,\frac{8P^*D}{\pi d^3} \tag{8-9}$$

where K is the Wahl correction factor that accounts for the effect of spring coil curvature on maximum stress (fig. 8-7). The spring index shown in figure 8-7 is the ratio of the mean coil diameter to the bar diameter, i.e.,

$$C = D/d \tag{8-10}$$

Equation 8-9 can be rearranged to give the load required to fully compress the spring in terms of the maximum allowable shear stress, i.e.,

$$p^* = \frac{\pi d^3}{8KD}\,\tau_{allow} = \frac{\pi d^2}{8KC}\,\tau_{allow} \tag{8-11}$$

Figure 8-8 gives τ_{allow} for two common spring steels as a function of bar and spring diameter.

5. The effective solid height of the spring is defined as the overall height of the active coils when fully compressed. The allowable deflection per unit length of effective solid height is

$$\delta = \frac{\Delta_t}{h} = \frac{\pi\tau_{allow}}{KG}\left[\frac{D}{d}\right]^2 = \frac{\pi C^2}{KG}\,\tau_{allow} \tag{8-12}$$

where h is the effective soild height. The number of active coils is

$$N = \frac{h}{d} \tag{8-13}$$

The total free height of the spring is the overall height under no-load conditions. It is given by

$$H = h + \Delta_t + nd \tag{8-14}$$

where n is the number of inactive coils.

6. In order to prevent lateral buckling of the spring when compressed, limits are placed on the ratio of free height to the mean coil diameter. Figure 8-9 shows the critical buckling ratio for two spring end conditions as a function of the ratio of total axial deflection to free height. Points below and to the left of the curves represent stable ratios. Points above and to the right of the curves represent ratios where lateral buckling is likely to occur unless spring guides are provided. Case A shown in figure 8-10 is not often encountered in practical applications and should be assumed only with careful judgment. Case B, which is generally applicable, can also be used with reasonable accuracy for springs with hinged ends, which are constrained to remain aligned axially (Harris-Crede, 1976).

7. The design process is a trial and error procedure involving the selection of a combination of spring properties that will provide the desired response without causing permanent deformation of the spring. Knowing the weight of the item to be shock isolated and the desired frequency of the isolation system, the required spring rate can be found from

$$k = \frac{4\pi^2}{g} Wf^2 \tag{8-15}$$

where

W = Weight of the item

f = Isolation system frequency, Hz

g = Gravitational constant

The static deflection is readily found from the relationship of equation 8-5. Unless specified otherwise, the dynamic deflection is generally taken to be equal to the static deflection. The total deflection to be accommodated can now be found by equation 8-7 after choosing a suitable value for C_f and the axial load, P^*, to fully compress the spring from equation 8-9.

8. At this point, one can choose a spring index, C, and find the corresponding Wahl correction factor, K, from figure 8-7. Then, with a suitable value for the allowable shear stress, the bar diameter can be found by solving equation 8-11 for d. The effective solid height, h, can be found from equation 8-12 and the number of active coils from equation 8-13. The total free height, H, from equation 8-14 should be checked for stability by means of figure 8-9. One would now have the required design parameters for a coil spring. The spring rate of the design spring should be calculated with equation 8-6 and compared with the desired system spring rate. If the comparison was not satisfactory, the process described above would be repeated until satisfactory agreement is obtained.

9. Helical coil springs are also required to resist lateral loads in some applications. In all cases, it is desirable to know how much resistance the spring offers to lateral displacement. For the case of a compression spring under vertical and lateral loads, with E = 30,000,000 psi (2.07×10^{11} N/m²) and G = 11,500,000 psi (7.93×10^{10} N/m²), Harris-Crede (1976) gives the lateral spring rate as

$$k_h = \frac{F_h}{\Delta_h} = \frac{d^4 \times 10^6}{C_h N D (0.204 H_s^2 + 0.265 D^2)} \text{ lb/in.} \tag{8-16}$$

$$= \frac{6.89 \times 10^7 d^4}{C_h N D (0.204 H_s^2 + 0.265 D^2)} \text{ N/m}$$

where

F_h = Lateral force

Δ_h = Lateral deflection due to F_h

C_h = Factor depending on Δ_{st}/H and H/D

H_s = Compressed height of spring ($h - \Delta_{st}$)

Values of C_h are given in figure 8-10. The ratio of axial to lateral spring rates for springs manufactured from materials where E/G $\simeq$ 2.6 (as above) is given by (Harris-Crede, 1976)

$$\frac{k}{k_h} = 1.44\, C_h \left[0.204 \left(\frac{H_s}{D} \right)^2 + 0.265 \right] \tag{8-17}$$

It has been found that lateral spring rates calculated from equation 8-16 may differ by as much as 25 percent from available test results. The differences are apparently due to deviations of the actual case from the idealized conditions assumed in the theory.

10. Tensile properties and torsional properties of typical helical coil spring materials are given in table 8-3.

(c) Torsion springs. Torsion spring isolators consist of a torsion bar to which dynamic loads are applied and a torsion-resistant member that resists the applied torque. The three basic types of torsion springs are: torsion bars, used for light to heavy loads; helical torsion springs, for light to moderate loads; and flat wire torsion springs, for light loads. A typical torsion spring is shown in figure 6-22.

1. Steels suggested for use in torsion bars include SAE 8660, SAE 9263. Table 8-4 shows the suggested maximum allowable shear stresses for these steels for various fabrication processes. The SAE (Society of Automotive Engineers) recommends a presetting strain of approximately 0.22 rad when loading is in one direction only. In applications where the loading can cause equal or nearly equal rotations in either direction of twist, presetting will not be beneficial.

2. The design criteria for a torsion bar isolator will normally include the weight to be supported, the frequency of the system (or spring rate for the bar), and an allowable deflection. If the weight and system frequency are given, the required spring rate in units of force per unit deflection can be determined from equation 8-15. The spring rate can be determined from equation 8-5 if the supported weight and allowable static deflection are specified. For small deflection angles, the spring rate of a torsion bar isolation system is given by

$$k = \frac{Gd^4}{10.2\, R^2 L_e} \quad (8\text{-}18)$$

where

G = Shear modulus for torsion bar material
d = Torsion bar diameter
R = Effective length of torsion lever
L_e = Effective length of torsion bar

The effective length of the torsion lever is the distance from the torsion bar center to the support strut center. The effective length of the torsion bar is that portion of the bar that is responsible for the bar's action.

3. It is usual practice to design a torsion bar system so that the lever arm is perpendicular to the support strut at the static position. Then the torque applied to the torsion bar can be reasonably approximated by

$$T \simeq RF \quad (8\text{-}19)$$

where F is the force at the end of the torsion lever. The length of the torsion lever must be determined either from specifications concerning a particular application or by a trial and error process. In the absence of other guidance, a lever arm length equal to twice the maximum deflection is suggested to give a nearly constant spring rate through the total deflection.

4. The force at the end of the torsion bar is equal to the product of the system spring rate and displacement. At total deflection, the maximum torque is given by

$$T^* \simeq Rk\Delta_t \quad (8\text{-}20)$$

and the torsion bar diameter required to resist the maximum torque can be found from

$$d = \left[\frac{16T^*}{\pi\tau_{allow}}\right]^{1/3} \quad (8\text{-}21)$$

5. The effective torsion bar length required to accommodate the maximum torque and total deflection is given by

$$L_e = \frac{\theta_t d^4 G}{584T^*} \quad (8\text{-}22)$$

where θ_t is the angular deflection in degrees associated with the total deflection obtained from equation 8-9, or

$$\theta_t = \sin^{-1}\left(C_f \frac{\Delta_{st}}{R}\right) + \sin^{-1}\left(C_f \frac{\Delta_{dyn}}{R}\right) \quad (8\text{-}23)$$

6. The portion of the bar with uniform diameter is a function of the effective length and the tapered length (fig. 6-23)

$$L_d = L_e - \frac{2}{3} L_t \left[\left(\frac{d}{d_o}\right) + \left(\frac{d}{d_o}\right)^2 + \left(\frac{d}{d_o}\right)^3\right] \quad (8\text{-}24)$$

where

d_o = Diameter of splined end
L_t = Length of bar taper

The length of bar taper depends upon the taper angle and the splined end diameter. The minimum splined end diameter recommended by the SAE is 1.2 d. The splined ends are faired into the basic bar with a 15 deg maximum taper angle and a fillet radius of 1.3 d (AJA, 1966). For a maximum taper angle and a minimum splined end diameter, the taper length is

$$L_t = 0.373\, d \quad (8\text{-}25)$$

7. The following procedure is suggested for selecting torsion bar isolator parameters. If the system frequency and supported mass are specified, use equation 8-15 to find the required spring rate. If the system weight and allowable static deflection are specified, the required spring rate can be determined from equation 8-5. Determine total deflection based upon the static deflection, required dynamic deflection, and factor of safety, using equation 8-7. Determine the maximum torque at total deflection from equation 8-20 using a specified or assumed torsion lever length. The required torsion bar diameter can then be obtained using equation 8-21 and an allowable shear stress. The allowable shear stress will depend on the fabrication process selected. The required effective bar length can be obtained from equation 8-22 after the total angular deflection has been determined from equation 8-23. The bar taper length can be computed after the splined end diameter is selected. If the minimal splined end diameter and maximum taper angle recommended by the Society of Automotive Engineers (SAE) are used, the tapered length can be determined from equation 8-25. Finally, the required uniform diameter bar length can be obtained from equation 8-24. The final design should be checked by computing the design spring rate using equation 8-18 and comparing it to the required spring rate.

(d) Disc springs (Belleville). Disc springs owe their high efficiency to a relatively uniform stress distribution that permits a high average utilization of the strength of the metal throughout the disc. Another contributing factor is that the critical stress is the compressive stress. Disc elements are typically "preset" to a total load and deformation that produces yielding in the area of maximum compression, and the residual stress in this area becomes tensile in the "no load" condition. It then becomes feasible to use a peak (imaginary) design compressive stress that is actually substantially beyond the yield point of the material; and the spring element, under load, behaves substantially as though the imaginary peak stress were real and were within the elastic limit. Finally, if additional yielding should occur at peak stress, there will be a very high rate of energy absorption at the expense of some small permanent reduction in the deformed height, h, of the disc. Under these loading conditions it appears feasible to use a maximum design stress, σ_m, of 450 ksi.

1. Other than design stress, the significant variables controlling disc spring element properties are the outside radius, R, thickness, t, and outside-to-inside diameter ratio, α. In order to minimize the number of isolator assemblies required, it is necessary to use the maximum feasible values of R and t for the disc elements. The need for heat treat hardness penetration to a substantial percentage of the thickness probably limits the maximum useful thickness to about 1.5 in. The disc diameter is limited to about 24 in., which is within the production equipment capacity limitations of known current disc spring manufacturers. Within the range of interest, spring element efficiency increases as the diameter ratio, α, decreases, and the disc element approaches the shape of a ring having a square cross section. However, at small values of ring width, W, the bearing line width becomes a substantial percentage of the total radial width of the element, and the actual (versus theoretical) properties of the spring elements become progressively less predictable. Consequently, a minimum width-to-thickness ratio of 2 should be used, and the minimum value of the diameter ratio is then

$$\alpha = R/(R - 2t) = 4/3 \qquad (8\text{-}26)$$

where R and t are the outer radius and thickness, respectively.

2. For disc spring elements of uniform thickness, the expressions for maximum spring force, F_m, and deflection, δ_m, can be simplified to (where Poisson's ratio is taken as 0.3)

$$F_m = \frac{2.991\, t^2 \sigma_m (\alpha^2 - 1)}{(\alpha^2 - 1) + 0.3715\alpha^2 \ln\alpha} \qquad (8\text{-}27)$$

and

$$\delta_m = \frac{1.6494}{Et} \sigma_m R^2 \left(\frac{(\alpha^2 - 1)^2 + 2.927\,\alpha^2 \ln\alpha}{[(\alpha^2 - 1) + 3.715\,\alpha^2 \ln]\alpha^2} \right) \qquad (8\text{-}28)$$

where E is the elastic modulus of the material (Wahl, 1963). Since the load/deflection curve is nearly linear at h/t values less than 1, the spring rate, k_s, and maximum energy storage capacity per unit weight, ξ_m, can be expressed as

$$k_s = \frac{1.8134\, Et^3}{R^2} \left(\frac{\alpha^4 - \alpha^2}{(\alpha^2 - 1) + 2.927\,\alpha^2 \ln^2\alpha} \right) \qquad (8\text{-}29)$$

and

$$\xi_m = \frac{0.7852}{E\rho_\omega} \sigma_m^2 \left(\frac{(\alpha^2 - 1)^2 + 2.927\,\alpha^2 \ln^2\alpha}{[(\alpha^2 - 1) + 3.715\,\alpha^2 \ln\alpha]^2} \right) \qquad (8\text{-}30)$$

where ρ_ω is the weight density of the metal. The very large potential advantage (about 5 to 1 based on ξ_m) of the disc spring element is apparent when these equations are compared to those for helical coil springs.

3. The basic equations for the properties of radially tapered disc springs are somewhat more unwieldy than those for disc springs of uniform thickness. However, for radially tapered springs having maximum deflection δ_m less than the material thickness, the equations may be simplified by introducing Poisson's ratio as a constant. Where Poisson's ratio is taken as 0.3, the equations of interest become (Wahl, 1963)

$$F_m = \sigma_m t_i^2/C_0 \qquad (8\text{-}31)$$

$$\delta_m = [0.5C_1 + (0.25\,C_1^2 + 0.01974)^{1/2}]^{1/3} + [0.5C_1 - (0.25C_1^2 + 0.01974)^{1/2}]^{1/3} \qquad (8\text{-}32)$$

$$k_e = \frac{E\, t_i\, (0.75\delta_m^2 + 1)}{C_2 R^2 (\alpha^2 + \alpha + 1)} \qquad (8\text{-}33)$$

and

$$\xi_m = \frac{0.2387\, E\, t_i\, \alpha^2 \delta_m^2\, (0.75\delta_m^2 + 1)}{C_2 R^4\, \rho_\omega\, (\alpha^3 - 1)(\alpha^2 + \alpha + 1)} \qquad (8\text{-}34)$$

where

$$C_0 = \frac{0.733\,[3.033(\alpha^{-0.5} - \alpha^{-1.533}) + 0.033(\alpha^{-1.533} - \alpha^{-0.5})]}{\alpha^{1.533} - \alpha^{-1.533}} - 0.1819$$

$$C_1 = \frac{0.667\, C_2 F_m R^2\, (\alpha^2 + \alpha + 1)}{E\, t_i}$$

$$C_2 = 0.8281\alpha^{-3} \left[\left(\frac{-1.7}{\alpha^{1.533} - \alpha^{-1.533}} \right) \left(\begin{array}{l} 2.907\,(2\alpha^{0.5} - \alpha^{-0.533} - \alpha^{-1.533}) + \\ 0.18\,(\alpha^{2.533} + \alpha^{-1.533} - 2\alpha^{0.5}) \end{array} \right) + \alpha - 1 \right]$$

k_e is the mean effective spring rate (integrated average of the nonlinear spring rate over the deflection range), t_i is the spring element thickness at the inner edge and the remaining symbols remain as previously defined. For an average thickness of 1.5 in. and $\alpha = 4/3$, t_i becomes 1.2857 in. and the numerical values of the above expressions become

F_m = 977,300 lb
d_m = 1.24 in.
k_e = 514,000 lb/in.
and
ξ_m = 4650 in. lb/lb

Compared to helical coil springs, the theoretical material reduction for radially tapered disc springs, based on energy storage at maximum stress, is nearly 90 percent. Compared to uniform thickness disc springs, the potential weight reduction is about 40 percent.

(e) Pneumatic springs. Pneumatic springs use a compressed gas as the elastic medium. They are relatively straightforward to design using standard seal components such as rolling diaphragms or bellows.

1. Pneumatic springs were used extensively within two buildings in the Safeguard system, the Perimeter Acquisition Radar Building and its Power Plant Building (Bradshaw-Sonnenburg, 1978). Where equipment weight was unknown, platforms were suspended from one ceiling by pneumatic isolators. Undamped pendulum motion provided isolation in the horizontal direction. A check of the isolators six months after initial adjustment showed a downward drift due to gas leakage. The maintenance policy for such isolators should call for frequent checking to prevent excessive platform settling. Furthermore, in adjusting the isolators, platforms can be raised so that null positions are at their upper limits. In this manner, a platform with pneumatic isolators may be allowed to settle through its entire tolerance range before servicing is required.

2. For a small change of volume as compared to the initial volume, and with the initial gas pressure sufficient to support the weight, or $P_oA = W$, the instantaneous frequency of the system is of the following form:

$$\omega^2 = \frac{nAg}{V} \qquad (8\text{-}35)$$

where
n = Gas constant for air
A = Piston area
V = Volume

3. The frequency is a constant and is a function only of piston area and initial volume. This is an important feature of the pneumatic isolator. If a platform with uneven distribution of weight is supported by pneumatic isolators, the platform may be leveled by pressurizing applicable isolators. This process shifts the vertical center of elastic supports to coincide with the center of gravity automatically without changing system frequencies. Thus, pneumatic isolators are able to minimize the coupling between the platform translational and rotational motions.

4. If the frequency of the system at static equilibrium is ω, $\Delta\omega/\omega$, the fractional increase in frequency from ω for a positive stroke of ζ inches, is as follows for an isothermal process:

$$\Delta\omega/\omega = \frac{\zeta\omega^2}{g - \zeta\omega^2} \qquad (8\text{-}36)$$

This relationship is plotted in figure 8-11.

5. For example, consider a shock isolation system that is designed for a maximum uncoupled vertical frequency of 1.2 Hz, with an expected maximum deflation of 8 in. As shown in figure 8-11, after a deflection of 8 in., a system originally at a frequency of 0.67 Hz will increase by 50 percent to approximately 1.01 Hz and a 0.8 Hz system would increase by 100 percent to 1.6 Hz. The required system would, therefore, be close to 0.7 Hz. Thus, the increase in frequency at 8-in. deflection would be by 73 percent (fig. 8-11) or 1.2 Hz and the stiffness ratio would be 3 (the square of 1.73) to give a displacement ratio of 0.4 in figure 6-6. This corresponds to an initial isolator gas chamber height of about 20 in. as may also be determined from the following equations. The height may be reduced by the use of an anxiliary gas chamber, as long as the total required initial gas volume is matched.

6. The spring rate of pneumatic cylinders is a function of displacement. At the neutral position, the spring rate of a single action cylinder is given by (fig. 6-6).

$$k_o = \frac{nAp_o}{L_o} \qquad (8\text{-}37)$$

where
L_o = Displacement
p_o = Cylinder chamber air pressure at static position

The gas constant n for air to be used in pneumatic spring computations is a function of temperature; however, for temperatures less than 500°F (260°C), it can be taken equal to 1.4 with only a slight error. At positions other than the neutral position, the spring rate can be determined from

$$k(y) = \frac{n\,A\,p'}{(L_o - y)} \qquad (8\text{-}38)$$

where p' is the cylinder pressure at the displaced position. Note that p' must be determined at each position of interest. The piston action is usually considered to be a quasistatic-adiabatic process (zero heat exchange), in which case the pressure can be determined from

$$p' = p_o \left[\frac{L_o}{L_o - y} \right]^n \qquad (8\text{-}39)$$

7. For double-action cylinders the neutral position spring rate is given by

$$k_o = n \left[\frac{p_{o1} A_1}{L_{o1}} + \frac{p_{o2} A_2}{L_{o2}} \right] \qquad (8\text{-}40)$$

Note that A_1 will differ from A_2 by the area of the piston shaft, although in most cases the difference is negligible. At positions other than the neutral position, the spring rate is given by

$$k(y) = n \left[\frac{p_1' A_1}{(L_{o1} - y)} + \frac{p_2' A_2}{(L_{o2} - y)} \right] \qquad (8\text{-}41)$$

As in the case of single-action pneumatic cylinders, the pressure at displaced positions will determine the spring rate. For a quasistatic-adiabatic process, the pressures are given by

$$p_1' = p_1 \left[\frac{L_{o1}}{L_{o1} - y} \right]^n \qquad (8\text{-}42)$$

$$p_2' = p_2 \left[\frac{L_{o2}}{L_{o2} + y} \right]^n \qquad (8\text{-}43)$$

8. The addition of surge tanks is illustrated in figures 8-12 and 8-13. However, in most field applications, surge tanks are not used. In the event they are used, surge tanks will provide damping in a pneumatic system. The effect of the surge tanks on the neutral spring rate depends on the frequency. The neutral-position spring rate for low-frequency displacement is obtained from the cylinder and surge chamber volumes and pressures at the neutral position. For a single-action cylinder with a low frequency, the spring rate for an arbitrary position of the piston is given by

$$k = \frac{n\,p'\,A^2}{V_c' + V_s} \qquad (8\text{-}44)$$

where

V_c' = Cylinder chamber volume at arbitrary position

V_s = Surge tank volume

At high frequency, the air flow between the cylinder and the surge tank is limited and the spring rates given by equations 8-37 and 8-38 can be used.

9. For double action cylinders with equal surge tank and cylinder volumes and system pressures on both sides of the piston, the neutral position spring rate for low frequency displacements is given by

$$k = \frac{2np_oA^2}{V_c + V_s} \qquad (8\text{-}45)$$

Equations 8-40 and 8-41 can be used to determine the neutral and displaced spring rates of a damped double-action pneumatic cylinder under high-frequency displacements. Harris-Crede (1976) contains additional guidance on the effect of surge tanks on pneumatic spring characteristics.

10. The area of the pneumatic cylinder piston is based upon the allowable pressures in the cylinder and the loads that must be carried under static and dynamic conditions. For single-action pneumatic cylinders, the required area is given by

$$A = \frac{k_o\,\Delta_t}{p_{allow}} \qquad (8\text{-}46)$$

where p_{allow} is the maximum allowable system pressure. The pressure that must be maintained at the static neutral position is

$$p_o = \frac{k_o\,\Delta_{st}}{A} \qquad (8\text{-}47)$$

The final dimension necessary to describe a single-action cylinder is the piston chamber length at the static position. If a dynamic displacement has not been specified, the chamber length can be found from

$$L_o = \frac{n\,A\,p_o}{k_o} \qquad (8\text{-}48)$$

If a dynamic displacement is specified, the chamber length is based upon an assumed linear response range, i.e., upon a displacement ratio less than 0.3, which leads to

$$L_o = 3.33\,\Delta_{dyn} \qquad (8\text{-}49)$$

11. The dimensioning of a double-action cylinder is more difficult because of the additional pressurized air chamber. The piston area can be estimated from equation 8-46, if an allowable pressure has been specified. In the case of double-action cylinders, p_{allow} is taken to be the allowable pressure differential between the two sides of the piston. In most cases, the area of the piston shaft can be neglected and the areas A_1 and A_2 considered equal for preliminary design. The length of the chambers can also be taken equal to that given by equation 8-49 for a trial design.

12. The pressures required at the neutral position in the two chambers can then be determined from

$$p_{o2} = \frac{k_o}{2A} \left[\frac{L_o}{n} - \Delta_{st} \right] \qquad (8\text{-}50)$$

$$p_{o1} = \frac{k_o\,\Delta_{st}}{A} + p_{o2} \qquad (8\text{-}51)$$

The above sizing of the double-action cylinder provides a preliminary design configuration that may have to be revised to obtain the desired combination of spring properties.

13. The following general procedure is suggested for sizing of pneumatic cylinders. If the system frequency and supported weight are specified, equation 8-15 can be used to find the required spring rate. The spring rate obtained is the neutral position spring rate. If the system weight and allowable static deflection are specified, the required neutral position spring rate can be approximated by equation 8-5. The total deflection to be accommodated is obtained using the specified dynamic deflection and a factor of safety. The maximum allowable air pressure is selected by considering pneumatic cylinder size restrictions and load requirements. At this point, the designer must decide whether a double-action or single-action cylinder is required. If a single-action cylinder is to be used, the required piston area is determined from equation 8-46 and the required neutral position pressure from equation 8-47. The cylinder chamber length is obtained from equation 8-48 or 8-49. Note that this is the length at the neutral position and must be greater than the total deflection to be accommodated.

14. If a double-action cylinder is required, the piston area is determined from equation 8-46 with p_{allow} taken as the allowable pressure differential between the two sides of the piston. The piston shaft area can be assumed to be negligible (i.e., $A = A_1 = A_2$). The piston chamber length obtained from equation 8-49 is assumed equal for both chambers. The neutral position pressures required in the two chambers are determined from equations 8-50 and 8-51.

15. The neutral and displaced spring rates for a single action undamped, pneumatic cylinder are determined from equations 8-37 and 8-38, respectively. The neutral spring rate for a double-action cylinder is obtained from equation 8-40. At other positions, the double action spring rate is obtained from equation 8-41.

16. Damping can be incorporated in the design by the addition of surge tanks, although this is a seldom-used option. If surge tanks are used, the neutral and displaced spring rates may be altered depending on the displaced piston position. The low-frequency spring rate of a damped single-action pneumatic cylinder is given by equation 8-44 and the high-frequency neutral position spring rate by equation 8-37. The low-frequency neutral position spring rate of a damped double-action cylinder with surge tank and chamber volume and pressures equal on both sides is given by equation 8-45. The high frequency spring rates are given by equation 8-40 for the neutral position and equation 8-41 for displaced positions. The low-frequency spring rate is a function of displaced piston position (effective pressure) while the high-frequency spring rate is a function of the piston at neutral position (field effects of the directed air are predominant).

(f) Liquid springs. Liquid springs are similar to pneumatic springs but use compressible liquids as the spring element. Because of the high bulk modulus of most liquids (approximately 200,000 psi), very high liquid pressures result when a small change of volumes occurs. Transient pressures of 50,000 psi are not uncommon. Attendant problems are seal leakage and friction under loading condition. Large thermal variations may also cause problems in large dimension or pressure changes.

1. The natural frequency of a simple liquid isolator can be determined as

$$\omega^2 = \frac{A^2 \beta_E g}{VW} \qquad (8\text{-}52)$$

where

V = Liquid volume

β_E = Effective bulk modulus of the liquid (EQ. 8.62 and fig 8-15)

W = Supported weight

A = Piston area

g = Acceleration of gravity

The spring rate of a liquid spring is essentially linear within the small variations of its bulk modulus as a function of pressure. However, since the weight term is in the frequency expression, the frequency cannot be independently adjusted as in the case of a pneumatic isolator.

2. The outstanding advantage of liquid springs is their small size and weight; their use in aircraft landing gear, where weight is at a premium, is justified on this basis.

3. Temperature changes in liquid spring fluids, due either to external influences or to internal heating resulting from load cycling, produce distinct changes in spring characteristics. Heating increases the preload pressure, or, under fixed load, changes the static load spring length and increases the spring rate. Temperature reduction has the opposite effect. In installations subject to wide variations in ambient temperature, thermostat-controlled heating blankets are sometimes used to maintain an approximately constant temperature near the upper end of the anticipated range.

4. Liquid spring cylinders are high-pressure vessels and are normally preloaded to several thousand psi whether or not a static load is supported. Consequently, the cylinder design should be analyzed thoroughly for all probable combinations of loading and stress. In general, the safety factor for all design stresses, based on the elastic limits of the material, should not be less than five.

5. For most liquid spring applications, the fluids having the best combination of desirable characteristics (high compressibility, temperature and chemical stability, and suitable viscosity) are the silicone oils. Of these, the most widely used are the General Electric Company's SF96 series, Union Carbide's L-45 series, Dow Corning's D.C. 210 series, and Dow Corning F-4029. All of these have slightly nonlinear stress/strain curves with the bulk moduli increasing with increasing pressure (fig. 8-15). It will be noted that the highest compressibility is available in the Dow Corning 210 0.65 centistoke fluid (fig. 6-8). However, the very low viscosity of this fluid may make sealing more difficult, and it is also very volatile compared to the higher viscosity fluids. The Dow Corning F-4029 fluid with nearly as low a bulk modulus and a viscosity of 100 centistoke is usually preferable.

6. The effective piston area of a liquid spring is taken equal to the net cross-section area of the piston rod. The required effective area is obtained from

$$A_e = \frac{k(y_s + y_d)}{p_{max}} \tag{8-53}$$

where

y_s = Static displacement (equilibrium position)
y_d = Maximum dynamic displacement
p_{max} = Maximum cylinder pressure
k = Spring rate

7. The static pressure required to maintain the spring at a specified static deflection is obtained from

$$p_s = \frac{k\,y_s}{A_e} + p_{p1} \tag{8-54}$$

where p_{p1} is the preload pressure (fluid pressure prior to any load being applied). Defining the cylinder volume under no applied load as V_o, the cylinder volumes at maximum static and dynamic deflections (neglecting cylinder breathing) are:

$$\begin{aligned} V_s &= V_o - A_e y_s \\ V_d &= V_o - A_e(y_s + y_d) \end{aligned} \tag{8-55}$$

The associated compressibilities are defined as

$$\begin{aligned} C_s &= \frac{V_o - V_s}{V_o} = \frac{A_e y_s}{V_o} \\ C_d &= \frac{V_o - V_d}{V_o} = \frac{A_e(y_s + y_d)}{V_o} \end{aligned} \tag{8-56}$$

Combining the above equations leads to the following expression for required cylinder volume under no-load conditions:

$$V_o = \frac{A_e\,y_d}{C_d - C_s} \tag{8-57}$$

Failure should be avoided if a working stress equal to 20 percent of the material yield stress is used in all computations.

8. The required piston rod diameter (fig. 8-14) is given by

$$D_r = 2\left[\frac{k(y_s + y_d)}{\pi\sigma_w}\right]^{1/2} \tag{8-58}$$

where σ_w is the allowable working stress for the piston rod material. The piston diameter that will provide the required effective area is obtained from

$$D_p = \left[\frac{4A_e}{\pi} + D_r^2\right]^{1/2} \tag{8-59}$$

9. The inside diameter of the cylinder is based on the required cylinder volume and an assumed cylinder length. Since the piston must be able to deflect in both directions, the cylinder length is usually taken to be slightly greater than twice the peak deflection. When a cylinder length has been assumed, the cylinder diameter is given by

$$D_c = \left[\frac{4}{\pi}\,\frac{V_o}{L_c} + D_r^2\right]^{1/2} \tag{8-60}$$

where L_c is the length of the cylinder. If the cylinder diameter proves to be excessively large, it may be reduced by selecting a longer cylinder length. The thickness of the cylinder wall can be obtained based on the maximum cylinder pressure and the allowable stress for the selected cylinder wall material. It will frequently be found that thin shell assumptions will not be appropriate in sizing the cylinder wall.

10. When the dimensions of the liquid spring have been determined, its actual spring rate can be estimated by

$$k = \frac{\beta_E A_e^2}{V} \tag{8-61}$$

where β_E is the effective bulk modulus of the fluid and V is the volume of the cylinder. The stiffness of a liquid spring varies proportionately with the load, which in turn is related to the displacement and cylinder volume.

11. Figure 8-15 shows typical bulk moduli for various fluids as a function of pressure. The effective bulk modulus can also be estimated by

$$\beta_E = \frac{V_o \Delta p}{\Delta V} \tag{8-62}$$

where V_o is the initial volume and ΔV is the change in volume caused by a pressure increment Δp.

12. The preliminary design procedure for a liquid spring is usually a series of trial and analysis steps until an acceptable configuration is obtained. The effects of temperature and change in cylinder volume due to interval pressures are usually neglected in preliminary design. Normally, static load, peak dynamic displacement, static displacement and maxi-

mum spring rate will be specified. When the cylinder fluid has been selected, preload and maximum cylinder pressures must be assumed. These pressures will be functions of the specified spring rates and deflections. The effective area of the piston is obtained from equation 8-53, using the assumed maximum cylinder pressure and specified values of the maximum spring rate and displacements. The pressure required to maintain the specified static deflection is given by equation 8-54. This is the equilibrium position under static loading. The compressibility of the cylinder fluid at static and maximum cylinder pressures is obtained from curves similar to figure 8-15 and the cylinder volume from equation 8-57. The physical dimensions of the piston rod, piston, and cylinder are obtained from equations 8-58, 8-59, and 8-60, respectively. As indicated previously, these latter steps require assuming a length of cylinder and several trials may be necessary to obtain the desired ratio of diameter to length. The spring rate at various deflections of interest can be obtained from equation 8-61.

(g) Dampers. The need for damping in the shock isolation system has been discussed in previous sections. Basically, dampers are required to fulfill the following functions:

- To limit the amplification of responses on top of the platform at low frequencies by damping out sustained oscillations of the shock isolation system at is natural frequencies
- To improve the system stability
- To reduce the rattlespace requirement

Dampers may be installed separately on the shock isolation system or they may be incorporated in the isolators as part of the elastic elements. Three types of damping are usually used: Coulomb or friction damping, orifice damping, and pure viscous damping.

1. The magnitude of the friction damping force is a constant independent of velocity or displacement, but has a direction that tends to oppose relative motion across the damper. The force is directly proportional to the coefficient of friction, the unit pressure and the area of contact. The friction damper is essentially a nonlinear device. It produces a dead band in the response motion, which may cause problems if the magnitude of the damping is large. Within the dead band, the friction damper transmits high-frequency, low-amplitude motions to the platform. These motions may not be acceptable because they cause large amplification of mounted equipment responses at high frequencies. Other problems are that friction factors are difficult to predict accurately and initial leveling of the platform may be hindered by large friction forces. Thus, even though friction dampers are relatively inexpensive to build, they should be considered only for applications where the damping force requirement is low.

2. When a fluid is pushed through an orifice restriction, a pressure drop is produced across the orifice. This pressure that acts on the piston pushing the fluid is the orifice damping force. The pressure drop across an orifice is a square function of the fluid velocity, which is in some proportion to the piston velocity. Thus, the orifice damping force is usually a square function of the piston velocity.

3. For an incompressible fluid, the damping force can be calculated as

$$F = \left[\frac{A^3 \alpha}{A_o^2 2g} \right] \dot{x}^2 \qquad (8\text{-}63)$$

where

A = Piston area
A_o = Orifice area
α = Fluid specific weight
$\dot{x}$ = Piston velocity
g = Acceleration of gravity

Note that the above expression has a constant coefficient.

4. For a compressible fluid, the damping equation is quite complicated, and no explicit form for the damping force can be written. The pressure behind the piston, p_u, can be related to the piston velocity by the following equation:

$$\left(\frac{\dot{x}A}{A_e}\right)^2 \left(\frac{k-1}{2gkRT}\right) = \left(\frac{p_d}{p_u}\right)^{2/k} - \left(\frac{p_d}{p_u}\right)^{(k+1)/k} \qquad (8\text{-}64)$$

where

R = Specific gas constant
T = Absolute gas temperature
k = Specific heat ratio
p_d = Outlet pressure at orifice
A_e = Effective area of orifice

The equation is highly nonlinear, and the solution of p_d as a function of $\dot{x}$ can be obtained only by a numerical iteration method.

5. The orifice type of damper can be easily designed into an isolator. However, because of the velocity-squared-to-damping force relationship, excessive damping may be introduced into the system. As a result, unacceptable accelerations may be transmitted to the platform. This situation is particularly likely when separate dampers are installed. For this reason, orifice damping should be used only when the input motion velocity is relatively low.

6. The simplest form of a viscous damper consists of two parallel plates separated by a film of viscous fluid. The shear force on the viscous damping force is

$$F = A\mu\dot{x}/y \qquad (8\text{-}65)$$

where

A = Plate area
μ = Absolute viscosity
$\dot{x}$ = Relative velocity of plates
y = Gap between plates

This type of damper is desirable because the damping force is a linear function of velocity. To produce sufficient damping force, a large plate area is required. This requirement may be difficult to meet by a design that can easily be incorporated into an isolator.

7. Another type of viscous damper makes use of the laminar flow property of the liquid. Consider a plug of length L and diameter D slides in a cylinder with a piston area of A. The gap between the plug and the cylinder is h. The viscous damping force is then

$$F = \left(\frac{12\mu LA^2}{\pi Dh^3}\right)\dot{x} \qquad (8\text{-}66)$$

where

$\dot{x}$ = Piston velocity
μ = Absolute viscosity

As the equation indicates, the damping force is also a linear function of velocity. The main difficulty is that for a large piston velocity, the gap has to be sufficiently small that the Reynold's number at the gap can meet the laminar flow requirement.

8. The damping requirement is usually specified in terms of a fraction of critical damping, ζ, for a single degree-of-freedom system with a viscous damper or

$$\zeta = \frac{C}{C_c} = \frac{C}{2m\omega_n} \qquad (8\text{-}67)$$

where

C = Damping coefficient
C_c = Critical damping coefficient
m = Mass
ω_n = Frequency

9. From the given damping ratio requirement, the damping forces can be calculated for different velocities. Then, different types of dampers can be designed to fulfill approximately the damping force requirements.

10. Figure 8-16 illustrates the sample design of a single acting, helical core spring shock isolator with linear viscous damper for tension loads only. The isolator is designed to support a dead load of 1000 lb at a spring rate of 102 lb/in., which corresponds to a frequency of 1 Hz. The designed stroke of ± 7 in. and the damping ratio is about 10 percent.

11. The purpose of this design is to convey certain general design principles and should not be construed as an optimum configuration. The points to note are the complete lack of hard noise paths through the isolator and the elimination of backlash and attendant impact between adjacent mechanical numbers. Of particular note is the elastomeric spring sleeve, teflon end pads, minimum clearance and low-friction piston rod guide and piston bearing ring. The choice of nonmetallic materials is arbitrary and in fact would depend on the detailed design. The piston bearing ring, for instance, might well be a relatively rigid elastomer such as Kel-F or rigid urethane rubber that has better wetting properties than teflon. The arrangement of the viscous damper as part of the piston reduces considerably the overall height of the isolator. The damping ratio may be varied by using liquid of different viscosities.

12. A second isolator design, also designed for a 1000-lb weight, but using compressed gas in lieu of a mechanical spring, is shown in figure 8-17. The isolator has a frequency range of 0.6 to 1.0 Hz for a stroke of ± 7.0 in. This pneumatic isolator design utilizes, for simplicity, an integral pneumatic damper operating in the pressurized air cavity. As with the mechanical isolator, no hard noise paths through the isolator exist, and friction and backlash are minimized. The elastomeric diaphragm is a rolling type and because of its stroke of ± 7 in. is nonstandard. Gas pressures are kept below 200 psi for diaphragm simplicity, low cost, and long life. Although the damper is nonlinear because of nonlaminar flow, it could be linearized for the anticipated 5 to 10 percent of critical damping. For high-frequency inputs, the damping chamber becomes the air support volume, and the diaphragm-to-piston volume becomes nonfunctional. Usually, however, displacements are sufficiently small at the higher frequencies that pressure perturbations are insignificant and this increase in natural frequency may be ignored. It is noted that for the same load-carrying capacity the pneumatic isolator is only about half the height of the mechanical spring isolator.

(3) Attachments

(a) The isolator attachment has the primary function of providing a flexible joint permitting three degrees of rotational freedom at the platform isolator interface. For most applications a joint is necessary to minimize bending loads to prevent isolator binding and to uncouple platform translational and rotational modes. An additional attachment function is to isolate the platform from high-frequency vibration inputs passing through or originating in the isolator.

(b) Spherical joints, conventional universal joints, and cables are some of the possible attachment designs. The specific choice depends on functionality and cost. A desirable design objective would be to

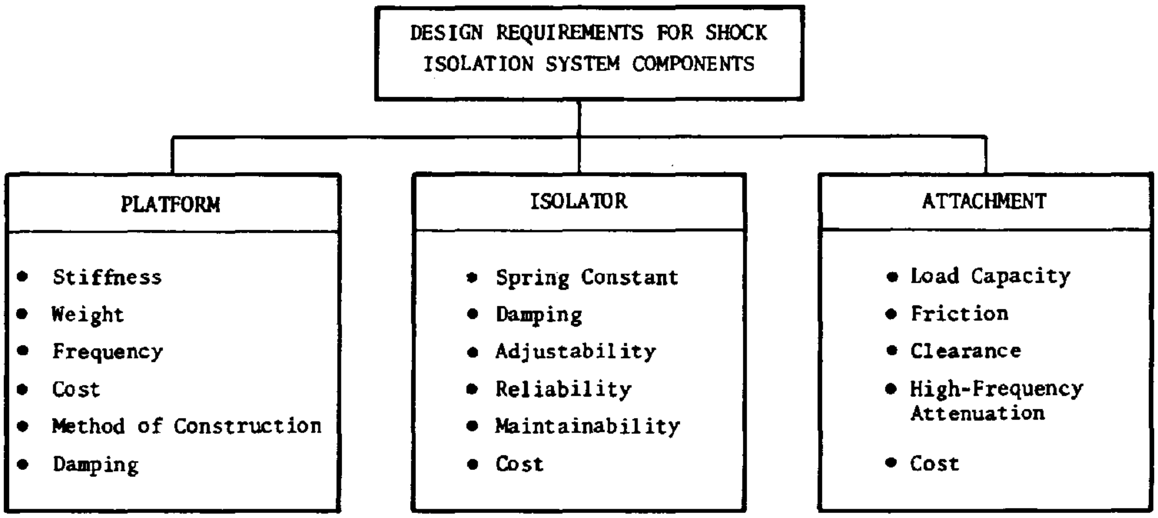


Figure 8-2. Summary of Shock Isolation System Design Requirements (Saffell, 1971)

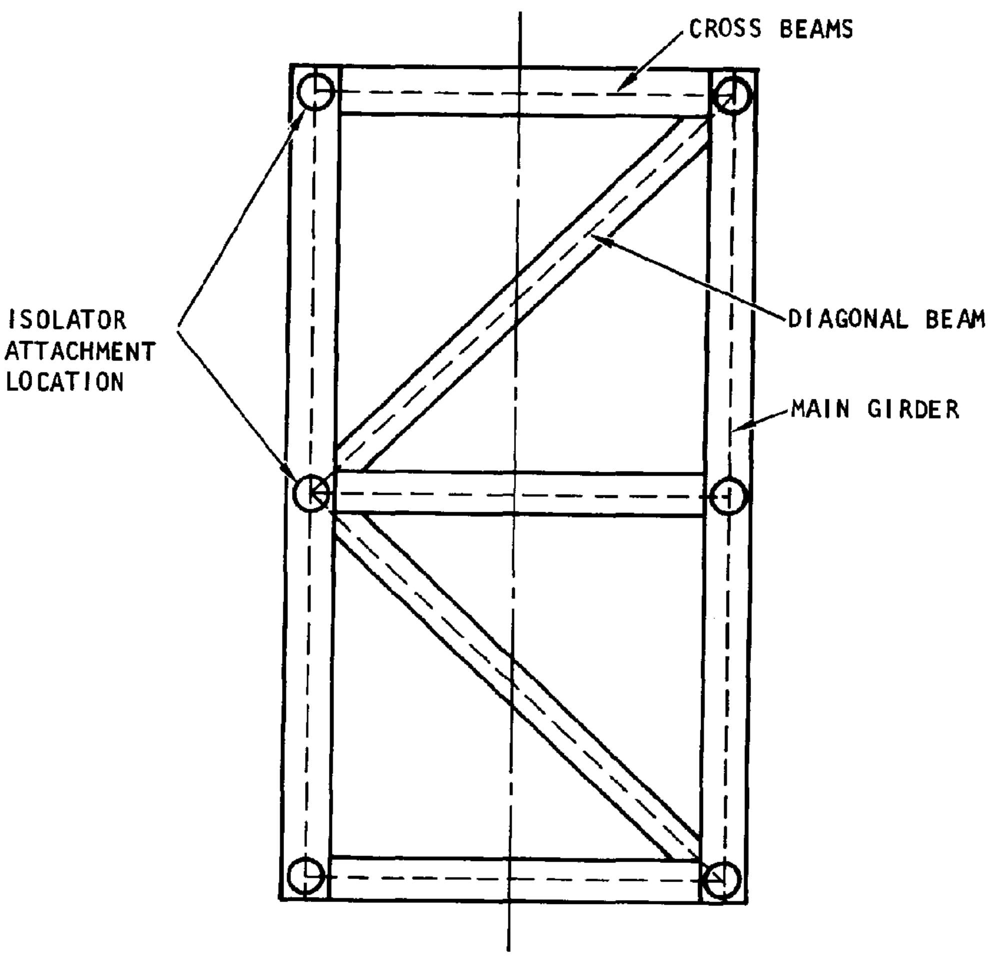


Figure 8-3. Diagonally Braced Frame Structure, First Platform Concept (Saffell, 1971)

minimize rotational dry friction, especially stiction. Dry friction causes several problems, the most significant being noise generation, isolator binding and bending, and modal coupling. Metal-to-metal impact, which might occur in connections permitting backlash or dead band, should be avoided.

(c) The typical design of attachments is shown in figures 8-16 and 8-17, the sample isolator designs. The end fittings shown are conventional spherical joints permitting three degrees of rotational freedom, two of the degrees of freedom being limited to approximately ±15 degree, assuming correct installation in the nominally centered position. The exact choice of bearing is unimportant as long as it satisfies the requirement of minimal backlash and friction and permits three degrees of rotation.

(d) Although the shock isolation system may be designed for relatively low frequencies, an impressive amount of empirical test data from existing shock-isolated facilities indicates that large amplitude, high-frequency motions may exist on the isolated platform. The addition of a highly damped secondary isolator, possibly integrally designed into the attachment, would probably be effective in suppressing the transmission of high-frequency motions. The choice of natural frequency of the secondary isolator should depend on the input frequency content, the damage criteria of the isolated equipment, the damping properties of the isolator, and the flexible body frequencies of the platform. Improper design of the secondary isolator could easily result in magnification rather than attenuation of motions if, for instance, natural frequency of the

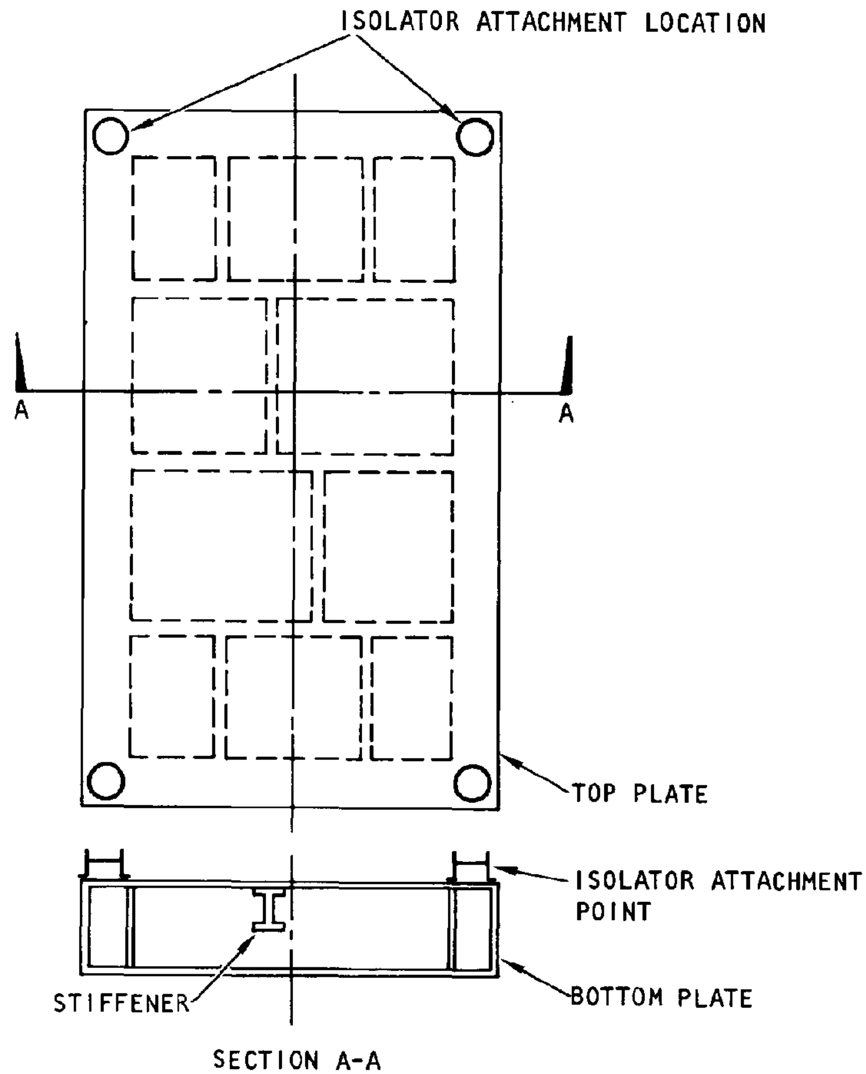


Figure 8-4. Plate Box Structure, Second Platform Concept (Saffell, 1971)

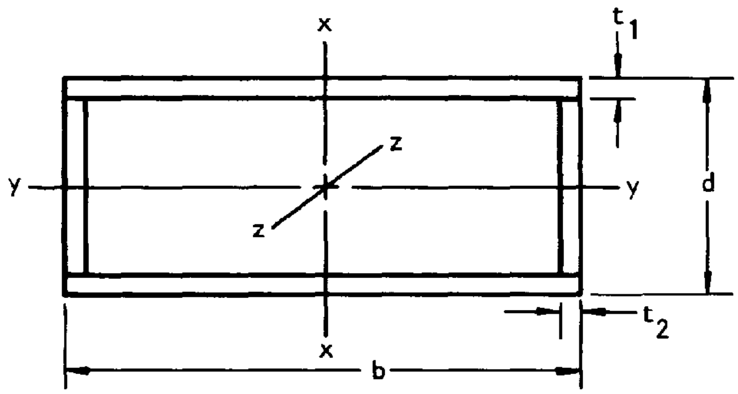


Figure 8-5. Approximate Platform Cross Section (Saffell, 1971)

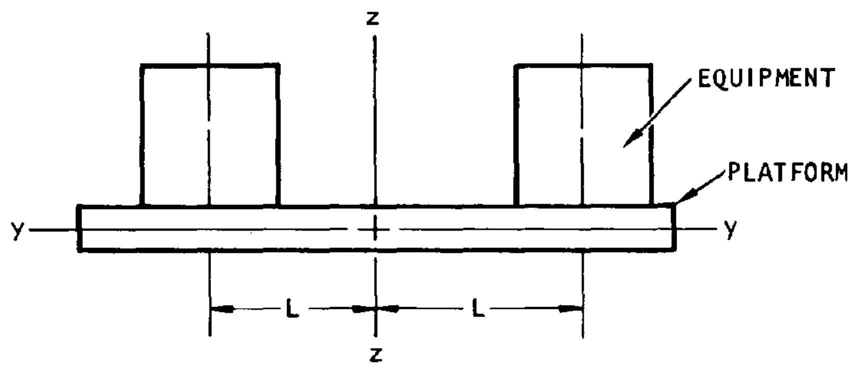


Figure 8-6. Assumed Platform and Equipment Arrangement (Saffell, 1971)

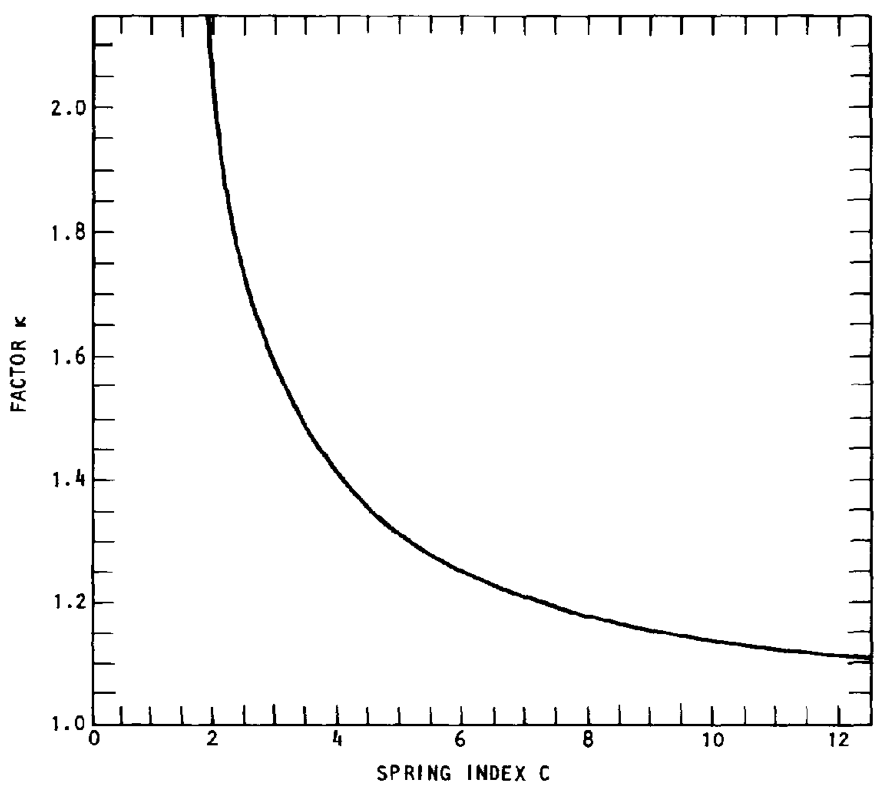


Figure 8-7. Values of Whal Stress Correction Factor for Round Helical Extension or Compression Springs (Saffell, 1971)

secondary isolator is close to an equipment natural frequency with high damage potential. It should be noted that amplification always occurs at resonant frequencies; large dampings are required to reduce this effect.

(e) Usually, elastomeric pads are employed as secondary isolators at the main shock isolator-to-platform attachment interface. As the ordinary elastomers give a damping ratio of about 15 percent, an amplification of about three to four over that of a rigid connection should be expected at a resonance condition. Thus, the successful application of a secondary isolator may depend on the use of a high damping material. Another approach is to establish first the frequency transmissibility of the main isolator and then select the frequency of the secondary isolator to match the

TABLE 8-2. COMPARISON OF DIFFERENT TYPES OF ELASTIC ELEMENTS (Saffell, 1971)

	Mechanical Spring	Pneumatic Spring	Hydraulic Spring	Elastomeric Spring
1. PERFORMANCE				
A. Frequency				
Stiffness	Helical, torsional and Belleville, variable pitch linear or nonlinear	Stiffness automatically changes with load to maintain constant frequency	Stiffness is nonlinear and strong function of temperature	Stiffness is nonlinear, but can be linearized for small percent deflection
Higher Harmonics	Surging and transients common	Almost never	Almost never	Limited
B. Damping				
Linear, Nonlinear	Negligible internal damping (0.0005); needs auxiliary damper	Easily included in design; usually nonlinear	Same as pneumatic	Inherent damping (0.05 to 0.2)
C. Adjustability				
Stiffness	Usually fixed by design	Easily varied	Fixed by design and fluid	None
Damping	None	Easily varied	Easily varied	None
Length	Usually difficult to adjust	Easily varied	Relatively fixed	None
D. Reliability				
Longevity	Excellent	Good	Fair	Good
Simplicity	Very simple	Simple	Complex	Very simple
E. Predictability				
	±10 percent	±1 percent	±5 percent	±15 percent
F. Envelope	Medium	Medium	Small	Medium
2. COST				
A. Engineering Cost				
Design	Low	Medium	Very high	Medium
Analysis	Low	Medium	High	Medium
Verification				
Laboratory	Low	Medium	Medium	Low
Quality Control	Low to medium	Low	Very high	Low
B. Fabrication Cost	Low to medium	Medium	Very expensive	Low
C. Installation Cost				
Initial	Low	Medium	Medium	Low
Tuning	High	Low	High	High
Verification	High	Low	High	High
Facility Interface	Low	Low	Low	Low
D. Maintenance Cost				
Inspection	Low	Medium	Medium	Low
Periodic displacement of components	Low	Medium	High	Low
Performance verification	Low	Low	High	Medium
3. MAINTAINABILITY				
Accessibility and Complexity	Virtually none required	Not as difficult as hydraulic, but periodic maintenance definitely required.	Difficult	None required, except possible degradation check.

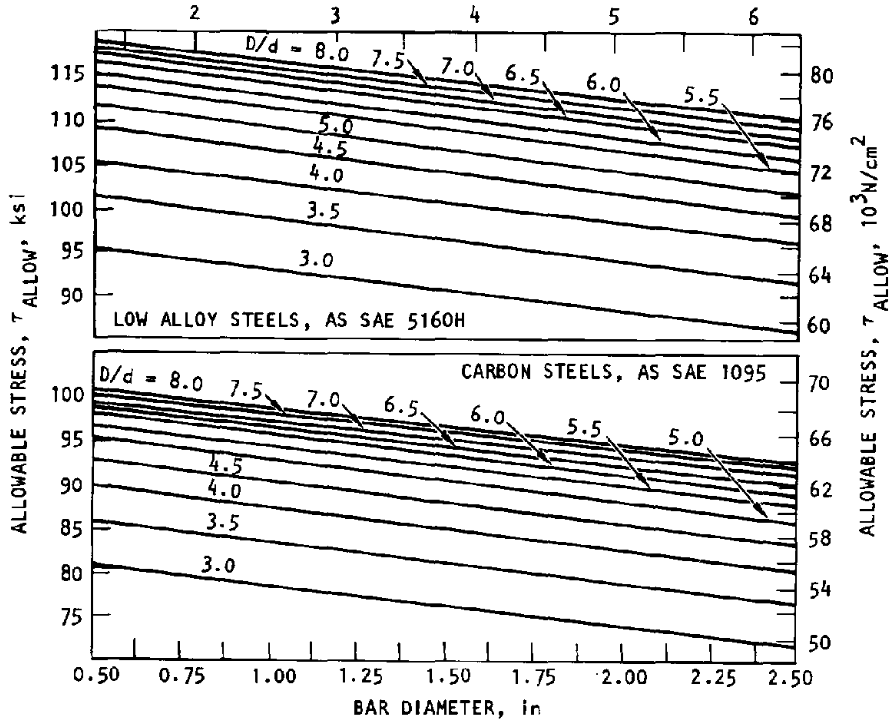


Figure 8-8. Recommended Maximum Stresses for Helical Springs Normally Subject to Dynamic Loading (AJA, 1966)

notch frequency of the established transmissibility curve.

(f) The elastic-pendulum type of shock isolation system requires a linkage from the ceiling to the isolator. Cables, rods, and tubes are some of the possible linkage designs. For a long linkage, the lateral frequency may be so low as to cause coupling effects with the shock isolation system frequencies. Thus, the lateral frequency of the linkage must be checked.

(g) For a rod or tube, the lowest lateral frequency occurs at the horizontal bending mode in the vertically unloaded condition. The frequency can be calculated as

$$f_n = \frac{\pi}{2L^2}\left[\frac{EIg}{A\alpha}\right]^{1/2} \qquad (8\text{-}68)$$

where

L = Linkage length
E = Modulus of elasticity
A = Cross-sectional area
α = Specific weight of material
I = Moment of inertia

From the equation, it is obvious that from frequency considerations, a large diameter, thin-wall tube should be used to maximize the I/A ratio.

(h) Cables have built-in damping, and because of their flexibility can be used directly without swivel joints. One significant disadvantage is their relatively low lateral frequency under loaded conditions. This low frequency permits a whipping action at close to suspension frequencies. The lateral frequency of a cable is a function of its stress, or

$$f_n = \frac{1}{2L}\left[\frac{Pg}{A\alpha}\right]^{1/2} \qquad (8\text{-}69)$$

where

P = Vertical load on cable
A = Cable cross-sectional area

(i) Another problem is that if cables are required to unwind or rotate with load changes, this may cause coupling of vertical motion with platform rotation.

8-4. Lessons learned from the Safeguard System, conclusions, and recommendations.

a. General. The following information has been taken from a report on the Safeguard Antiballistic Missile System Shock Isolation Systems (Bradshaw and Sonnenburg, 1978). This information pertains to the practical problems affecting the design, installation, and maintenance of shock isolation systems. Safeguard shock isolation systems included both helical spring isolators and pneumatic isolators.

b. Lessons learned.

(1) The amount of maintenance required for a particular platform was related to the platform size and the weight of equipment supported. The size influences the relative flexibility of the platform. For prac-

tical designs, the larger the platform the more difficult it becomes to obtain rigidity. Maximum platform rigidity is highly desirable in isolated systems for the following reasons:

(a) Natural frequencies of the platform-equipment configuration should be as high as possible above the fundamental frequencies of the isolation system, providing effective isolation from the floor environment.

(b) Superposition of non-rigid-body modes of vibration is minimal; this renders rigid platforms more amenable to simplified analysis.

(c) Routine maintenance problems involving adjustment of isolators and leveling of platforms are simplified on smaller, more rigid platforms because of the reduced number of isolators that must be adjusted simultaneously.

(2) The weight distribution factor appears strongly correlated with the frequency of servicing. Weight concentration reduces platform rigidity in a local region. An exaggerated case was found for a platform where the two large central isolators supported almost the total load, while the four end isolators were unable to support a significant portion of the load without ex-

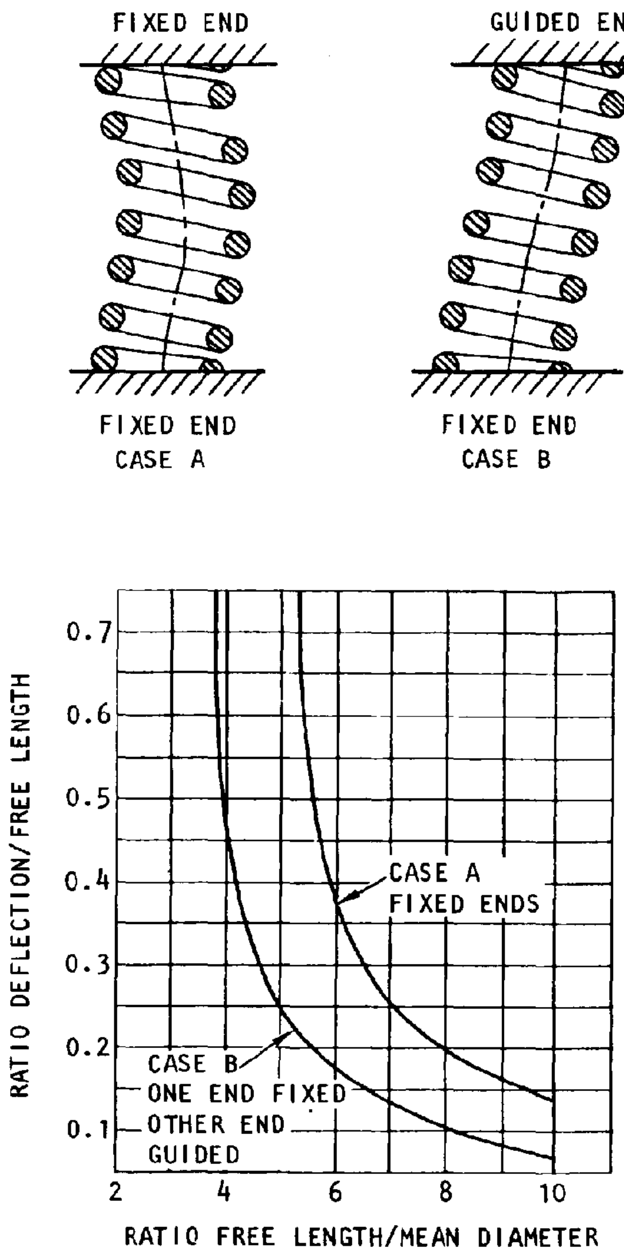


Figure 8-9. Critical Buckling Ratios (Wahl, 1963)

cessive platform bending. Uneven weight distribution may cause some isolators to settle rapidly while others may rise instead of settling. In either case, initial platform leveling is difficult to achieve and continuous servicing is required.

(3) All platforms were unlevel to some extent. There were no obvious cases where the unlevel condition caused a degradation of performance of the isolated equipment. The rather strict leveling requirements now in the maintenance instructions appear unnecessary, and may have more aesthetic than practical value.

(4) Initial platform adjustment should be referenced to the as-built extensions of platform structure or equipment. The as-built conditions are ignored by the present method of adjustment that uses the distance from the top of the platform to the floor.

(5) Minimum rattlespace was established from an envelope of the displacement floor response spectra for all locations at the perimeter acquisition radar (PAR) site. The maximum displacement obtained from shock spectra in this manner is applicable only to single-degree-of-freedom vibration systems. However, even the platforms that are small enough to behave approximately as rigid bodies have six degrees of freedom. This requires the superposition of responses (i.e.,

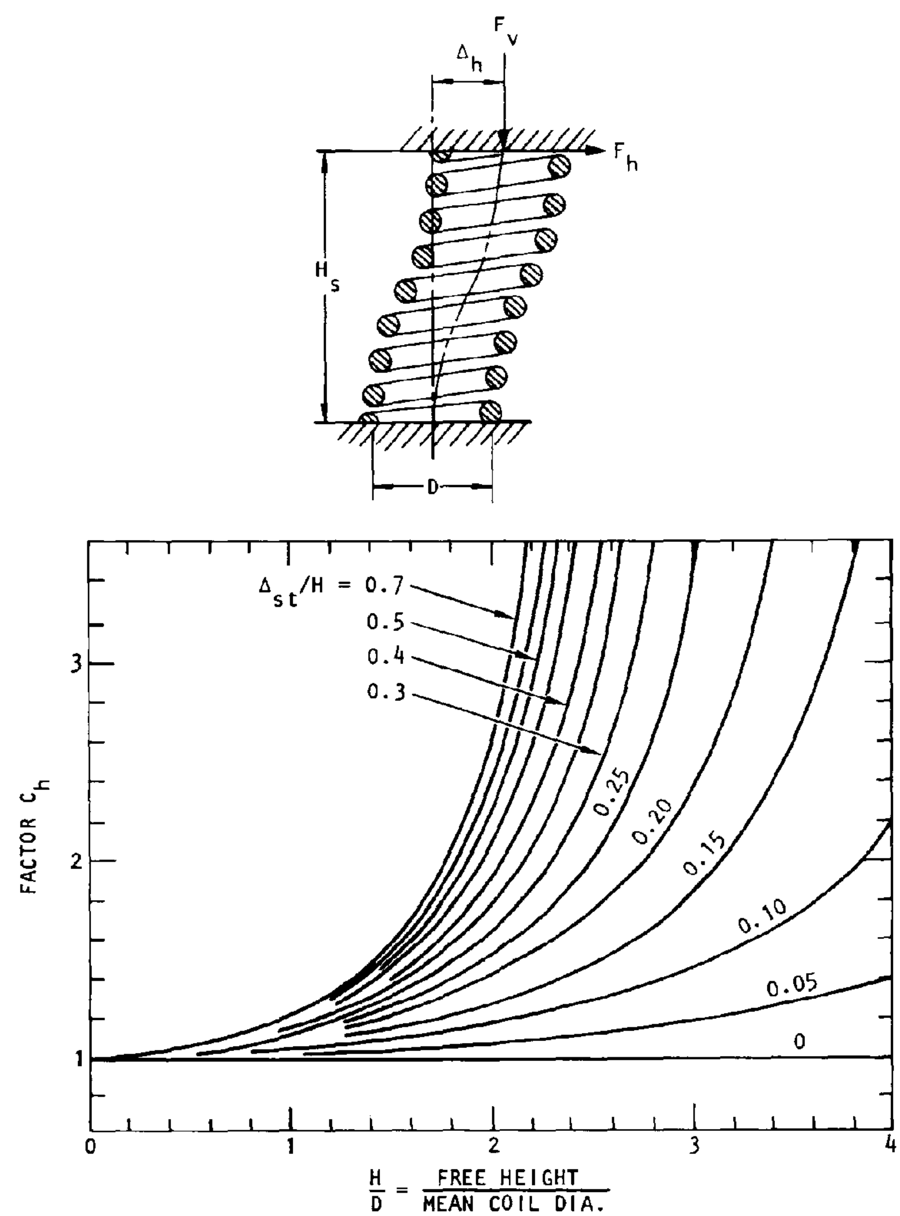


Figure 8-10. Lateral Stiffness of Helical Coil Spring (Harris-Crede, 1976)

displacement in this case) from each of the corresponding six modes of vibration. Use of the maximum displacement envelope of four inches for minimum rattlespace requirements is most likely sufficient, for engineering purposes, to waive the superposition effects for these platforms. This has never been demonstrated analytically, however.

(6) For the larger flexible platforms, there are many degrees of freedom to consider, and the superposition of modal displacements may be of greater engineering significance. The four-inch rattlespace criterion may not be conservative for these platforms. These platforms should be analyzed dynamically before alternative actions are considered. For example, some of the flexible platforms may be raised high enough to provide more rattlespace as needed, while others can be hardmounted to eliminate the response problem.

(7) If it can be demonstrated analytically that modal superposition is insignificant for the smaller

TABLE 8-3. TYPICAL HELICAL COIL SPRING MATERIALS (Harris-Crede, 1976)

Name and Composition	Tensile Properties*				Torsional Properties*			
	Ultimate Strength		Elastic Limit		Ultimate Strength		Elastic Limit	
	$10^3 lb/in^2$	$10^3 N/cm^2$	$10^3 lb/in^2$	$10^3 N/cm^2$	$10^3 lb/in^2$	$10^3 N/cm^2$	$10^3 lb/in^2$	$10^3 N/cm^2$
Carbon Steel SAE 1085	175-230	121-159	130-175	90-121	115-150	79-103	80-105	55-72
Carbon Steel SAE 1095	170-220	117-152	125-170	86-117	110-145	76-100	75-100	52-69
Alloy Steel SAE 4068	200-270	138-186	175-240	121-165	145-200	100-138	105-145	72-100
Chrome-Vanadium Alloy Steel SAE 6150	200-250	138-172	180-230	124-159	140-175	97-121	100-130	69-90
Chrome-Silicon Alloy Steel SAE 9254	250-325	172-224	220-300	152-207	160-200	100-138	130-160	90-110
Silicon-Manganese Alloy Steel SAE 9260	200-250	138-172	180-230	124-159	140-175	97-121	100-130	69-90

* Modulus of elasticity $E=30x10^6$ psi ($20.7x10^6 N/cm^2$); shear modulus $G=11.5x10^6$ psi ($7.93x10^6 N/cm^2$). Use slightly lower values of E and G for hot-wound springs.

TABLE 8-4. MAXIMUM ALLOWABLE SHEAR STRESS FOR TORSION BARS (AJA, 1966)

Allowable Stress		Fabrication Process
ksi	kN/cm^2	
140	96.5	Shot peened and preset
120	82.7	Preset
105	72.4	Shot peened
59 to 75	40.7 to 51.7	Unprocessed

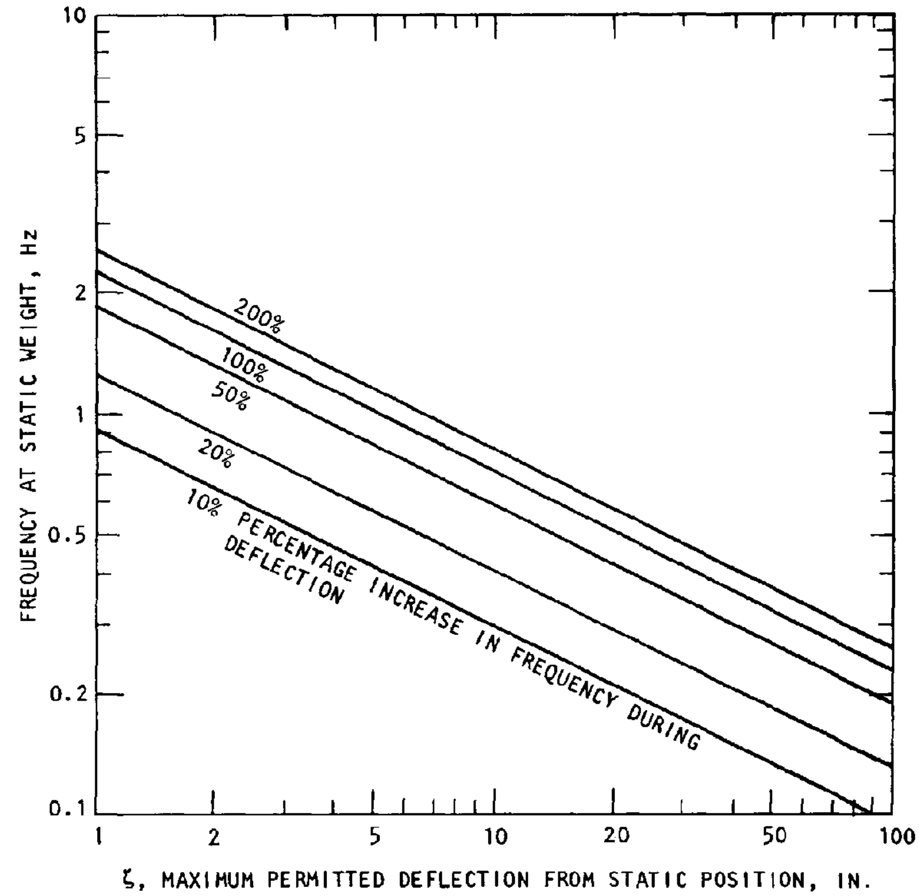


Figure 8-11. Fractional Increase in Frequency as a Function of Deflection and Frequency at Static Equilibrium (Saffell, 1971)

platforms, then a universal rattlespace criterion can be specified for these platforms. This analysis can be accomplished with a simplified linear rigid body computer program. The larger flexible platforms are more difficult to analyze in this respect and require a more comprehensive computer program. The rattlespace criteria for these platforms should be given individually, if superposition is found to be significant. The present minimum rattlespace criteria are incomplete.

(8) During this study, the maintenance rationale had to be deduced from original design and specification criteria. Platform modifications and isolator repairs are inevitable, and procedure have contradicted previous instructions. Before the maintenance instructions can be revised, the maintenance rationale must be reviewed and understood. Then, maintenance objectives should be documented in associated field instructions and future maintenance instructions should be compatible with the objectives.

(9) For the pendulum-type platforms installed, no distinction has been made between minimum and nominal rattlespace requirements. Minimum dimensions apply to horizontal displacement of the platform and its equipment. Nominal dimensions define vertical

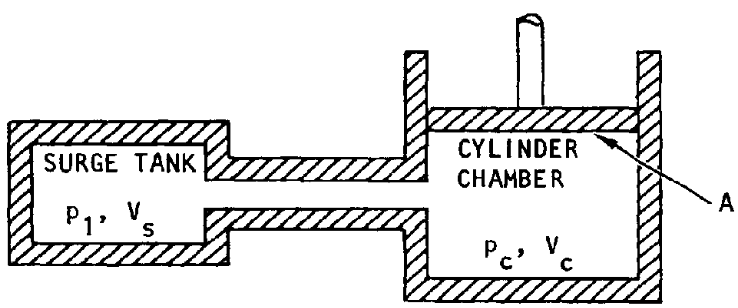


Figure 8-12. Damped Single-Action Pneumatic Cylinder (Crawford et at., 1974)

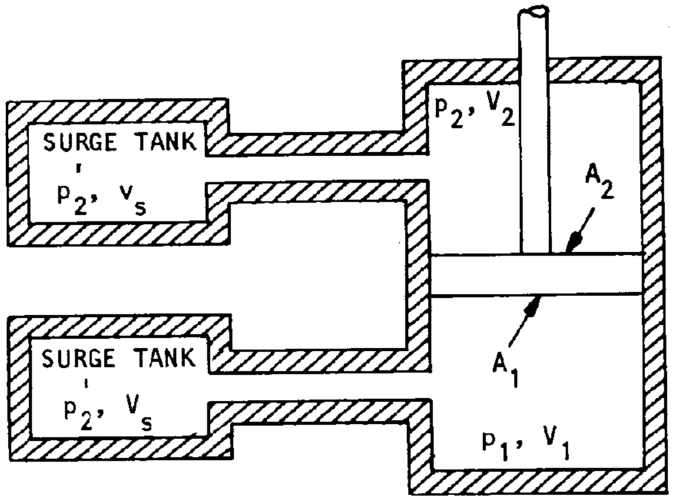


Figure 8-13. Damped Double-Action Pneumatic Cylinder (Crawford et al., 1974)

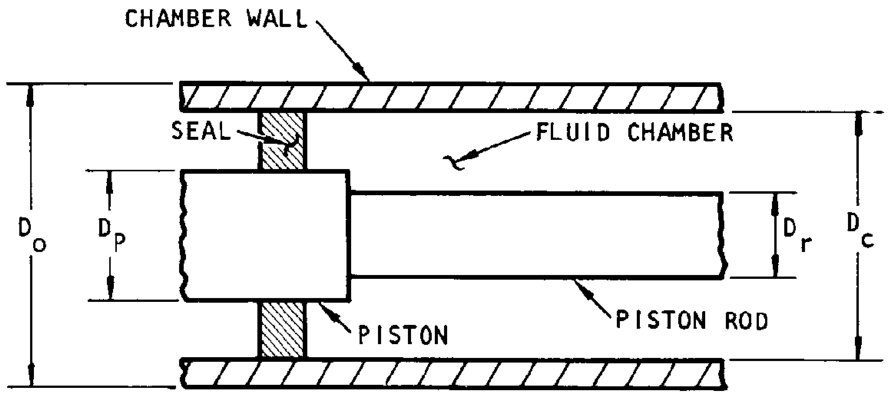


Figure 8-14. Tension Liquid Spring Schematic (Crawford et al., 1974)

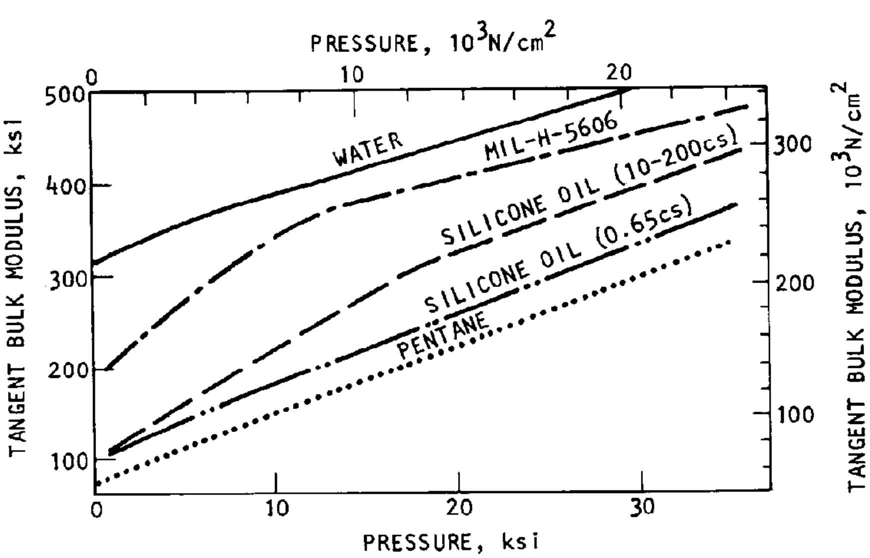


Figure 8-15. Bulk Modulus for Various Fluids (Crawford et al., 1974)

rattlespace together with allowable tolerances from the nominal positions. The nominal position plus or minus the tolerance must be established to satisfy the minimum vertical rattlespace requirement.

(10) In order to perform a dynamic analysis of a shock isolation system, the weight distribution must be known as well as the elastic and damping properties of the isolators. In the design stage, the weight information was not available for many of the platforms. Although these platforms might have been equipped with helical spring isolators, pneumatic isolators were used because of their capability to adjust pressure to support the unknown loads. Pneumatic isolators are convenient for this purpose, but continuous maintenance is necessary because of gas leakage. Based on Safeguard experience, future systems should not use pneumatic isolators unless weight distribution is unknown. For example, a better approach might be to design a platform for specific conditions (weight and c.g.) that exceed those expected. After the equipment is installed, weight can be added at the proper locations to bring the weight and c.g. in line with the design. Using

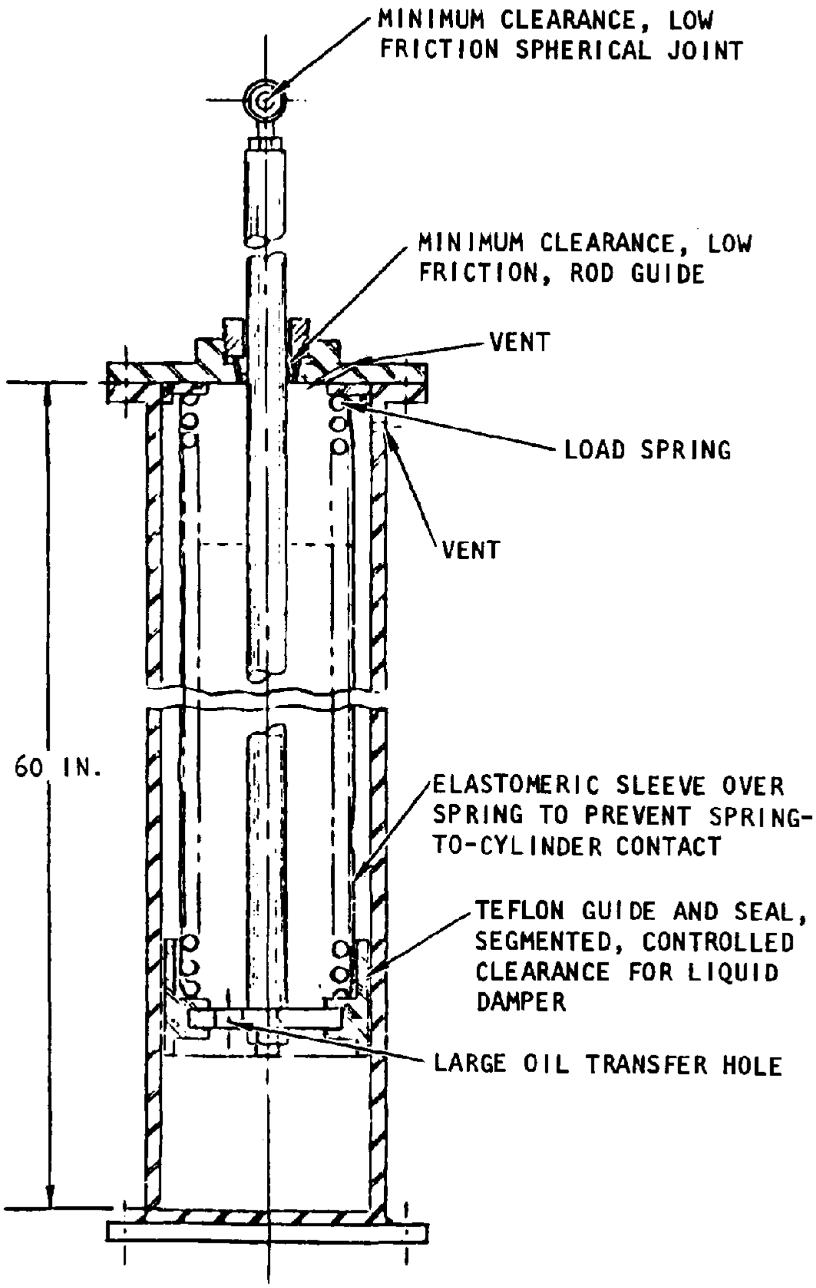


Figure 8-16. Single-Acting Helical Spring Shock Isolator with Linear Liquid Damping, Tension Loads Only (Saffell, 1971)

this approach, helical spring isolators could be used and future maintenance would be significantly reduced.

(11) Early in the Safeguard project, attempts were made to analyze the response properties for a few systems by using a comprehensive nonlinear shock isolation computer program (ISOL). Later, this program was increased in generality, and in its present form appears capable of treating flexible platforms as well as nonlinear isolators. Even though ISOL may now be a good shock isolation system design tool, it did not meet the needs of designers at that time. Thus little information is available from the analytical results. For the PAR site alone, 87 percent of the platforms involving 66 percent of the total isolators could have been analyzed both statically and dynamically using a linear rigid-body program with simplified forcing functions. It is noted that all of the isolators were designed to operate in their linear ranges, but 13 percent of the platforms appeared sufficiently flexible to require special consideration. It would have been appropriate to use ISOL for these flexible platforms, but not at the price of losing analytical support for the more rigid platforms because of the complexity of the program. Since the equipment weights were unknown, the designers needed a simplified linear rigid-body program, which they could operate without help, in order to observe the effects of changing and shifting weights within allowable limits. Furthermore, in accordance with the above paragraph, some (if not all) of the larger flexible platforms might have been divided into smaller units to achieve greater rigidity. These plat-

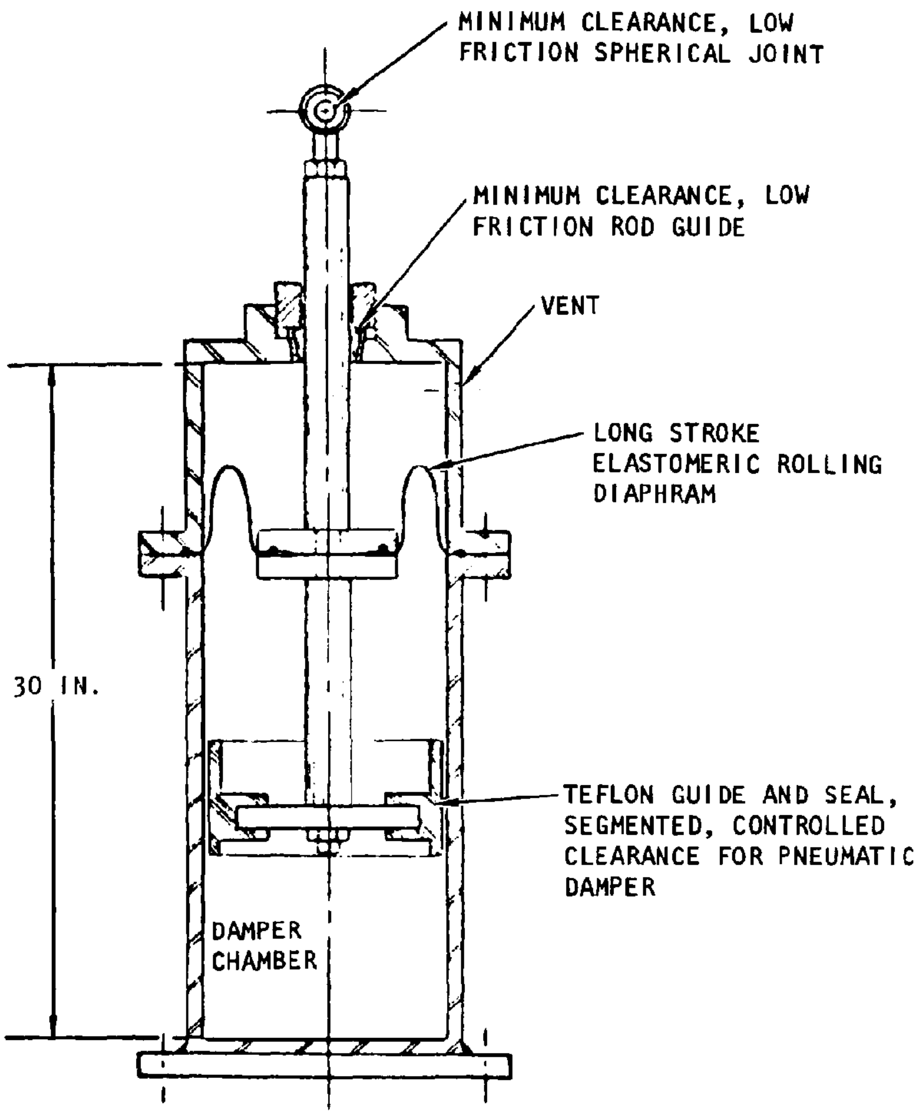


Figure 8-17. Single-Acting Pneumatic Shock Isolator with Pneumatic Damper, Compression Loads Only (Saffell, 1971)

forms could have also been analyzed by the simplified program. (The features of ISOL are summarized in appendix B.)

(12) Running machinery on a platform can cause a serious steady-state vibration problem that is unrelated to the threat shock environment or the isolators that support the platform. Isolation systems that are well designed will have mounted platform natural frequencies removed as far as possible above the fundamental frequencies removed as far as possible above the fundamental frequencies of the platform-isolator configuration. It is supposed that the fundamental system frequencies may not exceed 3 Hz, while the nonrigid mounted platform frequencies may range from 10 Hz to 500 Hz. With these conditions, it is not possible for frequencies above about 2 Hz from the shock environment to reach the sensitive equipment. However, alarming resonances can be generated on the mounted platform in the higher frequencies by running machinery. This phenomenon can be easily demonstrated, using a small shaker located any place (except at a node point) on the platform or equipment. Running machinery is never perfectly balanced and acts as a shaker at the running frequency. In one case at the PAR site, two generators were mounted on the same platform. When one generator was running, its rotating frequency coincided with the natural frequency or the other causing significant bearing damage. It is often possible to eliminate these problems by stiffening, shifting weights, or increasing effective damping in a local area. For complicated platform-equipment configurations, such resonances may be difficult to identify in the design stage by analytical methods. In cases where running equipment is relatively fragile and must be isolated, attempts should be made to analyze steady-state vibrations and avoid resonances by separating natural frequencies. These platforms should then be tested after installation and changes made if necessary.

(13) There are numerous platforms that have breakable drain pipes extending into the rattlespace below the platform. During the six-month interval between inspections, gas leakage from the pneumatic isolators causes the platforms to settle. If the platforms settle down to the isolator's lower tolerance limits, the pipes would be crushed. Because maintenance instructions have failed to address as-built conditions, maintenance personnel improvised by blocking the platforms, thus eliminating isolation. The use of breakable pipes is a poor design concept and can cause periodic down time for replacement. (Note: If an actual shock occurred, these pipes would be severed without sacrificing isolation system performance.)

c. Conclusions.

(1) The PAR site is not hardened to the expected shock environment in its present status.

(2) Helical spring isolators should have been carefully matched for loading when initially installed; such isolators require almost no maintenance after installation. The crude but adequate damping adjustment on these isolators should be checked periodically. (The maintenance required for damping adjustment is minimal.)

(3) Pneumatic isolators leak gas continuously and require periodic servicing, which involves platform adjustment. Servicing requirements are more frequent for those platforms with an uneven distribution of weight and for the larger flexible platforms with more than four isolators. With existing maintenance procedures, the Shreader valves on these isolators cause a significant maintenance problem because they will not permit accurate pressure adjustment. This problem should be insignificant if the recommended maintenance procedures are adopted.

(4) Existing periodic maintenance instructions are inadequate for the following reasons:

(a) In some cases, platform adjustment instructions are inconsistent with minimum rattlespace requirements.

(b) As-built rattlespace conditions have never been documented.

(c) Platform weight distributions have never been tabulated for ready reference. This information is necessary whenever static or dynamic analysis is required. An analysis should be made whenever possible and economical, such as during initial platform design and any time thereafter upon changing isolator capacities or weight distribution.

(d) The existing semiannual isolator inspection and platform adjustment requirement is too frequent in some cases, and too infrequent in others.

d. Recommendations. Assuming that hardness for the PAR site must be achieved and maintained, the following recommendations are proposed:

(1) Rewrite the maintenance instructions to use a walk-through inspection.

(2) Perform a detailed rattlespace survey and update existing drawings and documents accordingly.

(3) Use the results of the rattlespace survey to mark upper and lower null deviation limits on the isolator shafts. This will permit full exploitation of the walk-through inspection procedure and minimize periodic maintenance.

CHAPTER 9

VALIDATION

9-1. Introduction.

a. The validation of a shock isolation system involves testing and analysis to determine the probability that with at least 90 percent confidence the system will meet mission requirements during and after an attack.

b. A validation test program may be composed of all or part of the following:

—Equipment environmental tests (qualification)
—Isolator tests
—Twang tests
—Impedance and transfer function measurements
—Quality control tests
—Full-scale platform dynamic tests
—In-place pulse simulation tests
—Facility tests

This chapter discusses these tests and measurements and shows how they are used in validation.

9-2. Element and component validation.

a. Equipment tests.

(1) Traditionally, equipment located in protective facilities has been subjected to both environmental qualification tests for prototypes and production or quality control environmental tests. Production-line environmental tests (PET) are applied to equipment destined for installation in protective facilities. The tests are used to screen out defects in workmanship and material.

(2) The level of environmental severity used for quality control tests can be the same as or lower than that used for qualification tests. Qualification tests measure the performance of the equipment design to meet expected shock environments and normally include higher test levels to account for limitation of testing, environmental uncertainty, and statistical variation in the sample equipment tested.

(3) Equipment tests are typically conducted using shock machines and vibration shakers in test laboratories. Vibration tests have been conducted on both single and biaxial machines. Figure 9-1 illustrates a typical test setup and is a schematic of the biaxial vibration test machine at the U.S. Army Construction Engineering Research Laboratory at Champaign, Illinois. Table 9-1 lists the major large-size vibration test facilities in the United States in addition to the facilities of the U.S. Army at Champaign. A large number of electrodynamic vibration machines are also available and are located in numerous U.S. Government laboratories, Aerospace Corporations, and independent testing laboratories.

(4) An extensive discussion of equipment testing for fragility, as well as of qualification and production environmental tests, has been provided in chapter 4. Additionally, test results for different types of tests and for a wide range of equipments are provided in appendix A of this volume.

b. Isolator tests.

(1) Isolator tests are usually conducted on individual isolators. Static compression and extension tests should be made to deflection limits required in the procurement specifications. Spring constants should be determined to establish that measured values are within specification tolerances.

(2) Dynamic tests can be performed as illustrated in figures 9-2, 9-3, and 9-4. The excitation for these dynamic tests was provided by pneumatic and air/hydraulic actuators capable of high rates of loading. The data from these tests were analyzed to assess the performance characteristics of the isolators for a range of loads and to determine compliance to specifications. Dynamic tests such as these are also useful for measuring and evaluating the stress levels of fittings, attachments, swivel joints, and other hardware, as well as for the onset of potential failure.

(3) In some cases simple tests may be performed, such as a pull-down of the test weight with a subsequent quick release (twang test). Another variation is to drop the entire isolator with the test weight and let the opposite end of the isolator impact on a bracket to produce a velocity shock by sudden arrest from the free fall condition.

9-3. Dynamic measurement and response prediction.

a. Transfer (system) functions.

(1) To improve response-prediction accuracy requires inplace measurements of the dynamic properties of a shock-isolated system. The structure's transfer function must be measured from building/isolator attachment points to several points near equipment mounts on the shock-isolated platforms. The platform response motions may then be calculated by convolving these structure transfer functions with the building shock motions that would be induced by nuclear attack.

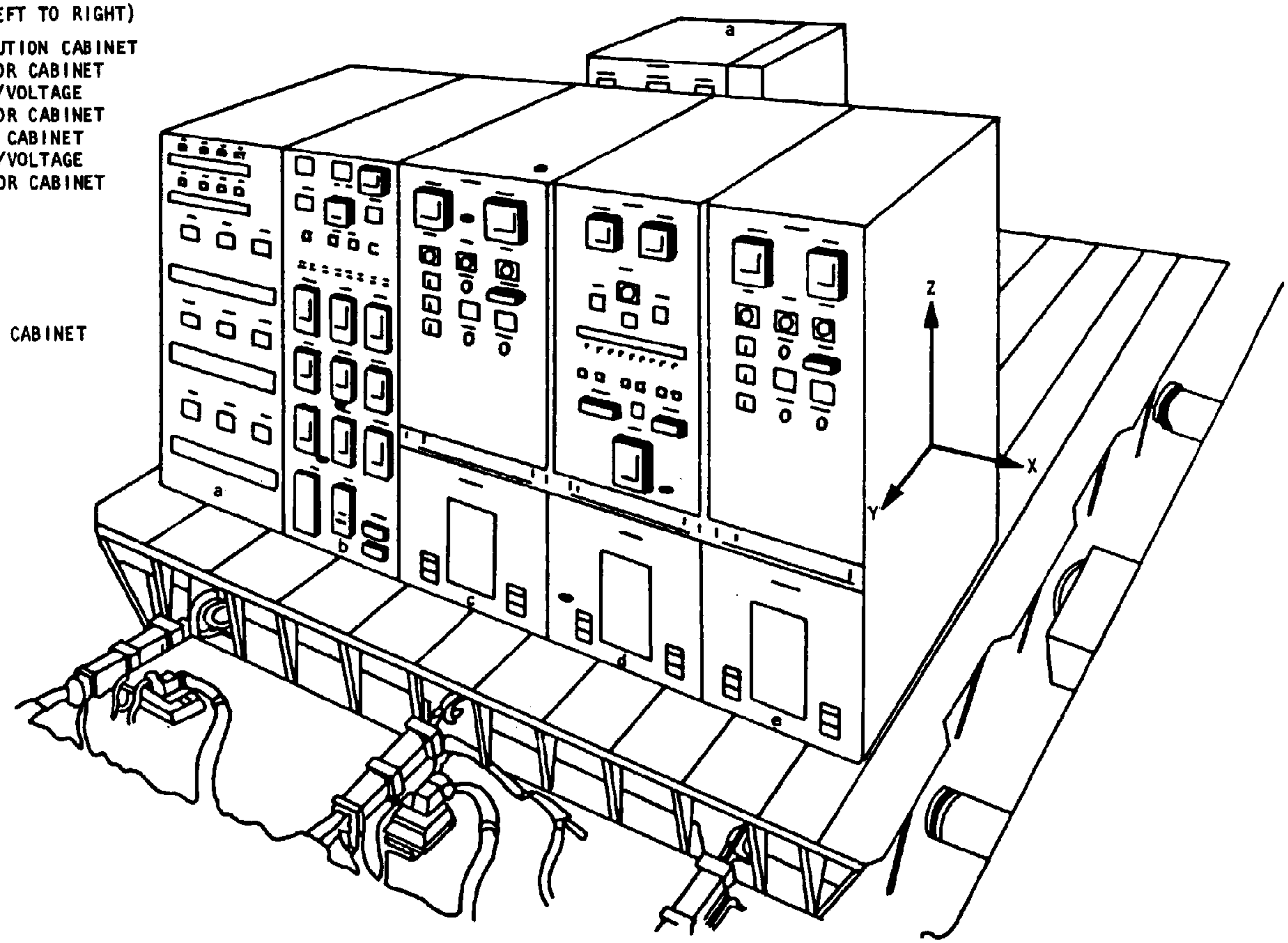


Figure 9-1. Qualification Test Setup for Motor Generator Set Mounted on on Biaxial Electrohydraulic Vibration Shaker. (Programmed Two Axes Transient Vibration Closely Approximated Threat Motion) (COE, 1974)

TABLE 9-1. MAJOR U.S. VIBRATION TEST FACILITIES (AA, 1978c)

				Maximum Table Input Motions (Transient)			
Organization/Location	Machine (Facility) Designation	Maximum Specimen Dimensions L by W by H, ft	Maximum Specimen Weight (g-limit), lb	Acceleration, g	Velocity, in./sec	Displacement (Double Amplitude), in.	Mode of Operation
Wyle Laboratories Huntsville, AL	Machine C	17 ft by 11 ft by 40 ft	60,000 (2 g)	8	35	6	Simultaneous Biaxial
	Machine D	12 ft by 12 ft by 12 ft	10,000 (1.5 g)	3	25	8	Simultaneous Biaxial
Wyle Laboratories Norco, CA	Machine G	8 ft by 12 ft by 12 ft	10,000 (2 g)	7 (h) 8 (v)	46 (h) 33 (v)	12 (h) 9 (v)	Simultaneous Biaxial
	Machine B	8 ft by 8 ft by 12 ft	12,000 (2 g)	12	30	3	Single Axis (horizontal or vertical)
U.S. Bureau of Reclamation Denver, CO	USBR Vibration Test System	20 ft* by 15 ft* by 10 ft*	10,000*	3*	30	10	Single Axis (horizontal)
U.S. Army Construction Engineering Research Laboratory Champaign, IL	CERL Biaxial Shock-Test Machine	12 ft by 12 ft by 12 ft*	15,000 (30 g)	20* (h) 30* (v)	32 (h) 27 (v)	6 (h) 3 (v)	Simultaneous Biaxial
University of California Berkeley, CA	Earthquake Engineering Research Center Earthquake Simulator Laboratory	20 ft by 20 ft by 40 ft	100,000 (1.0 g)	1.5 (h) 1.0 (v)	25 (h) 15 (v)	10 (h) 4 (v)	Simultaneous Biaxial
University of Illinois Urbana, IL	Structural Research Laboratory Earthquake Simulator	12 ft by 12 ft by 40 ft*	10,000	7.5	15	4	Single Axis (horizontal)
Westinghouse Advanced Energy Systems Division Large, PA	Seismic Test Laboratory	40 ft by 12 ft by 30 ft	180,000	0.3	--	20	Single Axis (horizontal)
		8 ft by 12 ft by 30 ft	40,000	1.4	--	20	Single Axis (vertical)
		9 ft by 9 ft by 30 ft	40,000	1.4	--	20	Vector Biaxial (air bag suspension-simultaneous biaxial)
Applied Nucleonics Company Santa Monica, CA	Cable Tray and Conduit Raceway Test Fixture	40 ft by 13 ft by 13 ft	10,000	1.2	30	6	Vector Biaxial
Approved Engineering Test Labs Los Angeles, CA	System 2	6 ft by 6 ft by 12 ft*	10,000	6	30*	12	Simultaneous Biaxial

*Estimated Value h: Horizontal v: Vertical

(2) Both the in-place tests of platforms and the calculated motions from transfer functions measured in-place provide additional data for the verification of equipment performance under specified threat. These results may be compared with the prior laboratory vibration tests of individual units to determine the degree to which the equipment has met or exceeded hardness requirements (Safford-Walker, 1975; Safford et al., 1977b).

(3) Physical measurement techniques are applied to obtain the mechanical impedance properties of a structure. Selected points on the structure are excited over the frequency range of interest. The structure's response to the vibratory force is quantitatively measured at both the drive point and other selected points. By using the Fourier transformation, it is then possible to obtain the ratio of the output (force or motion) to an input (force or motion). However, the usefulness of this technique is limited to a structure that is linear or nearly linear so that the principles of superposition will apply. The method is indicated by equation 9-1. Force ratio per drive point:

$$\tau_F(\omega) = \frac{F_o(\omega)}{F_i(\omega)} = \frac{|F_o(\omega)|e^{j\theta(\omega)}}{|F_i(\omega)|e^{j\phi(\omega)}} \qquad (9\text{-}1)$$

where

$F_o(\omega)$ = Complex output force at isolator/building junction

$F_i(\omega)$ = Complex input force at drive point

θ,ϕ = Phase angles

(4) An indirect method of transfer function measurement can be used to avoid the impracticality of inserting a motion generator between a massive piece of

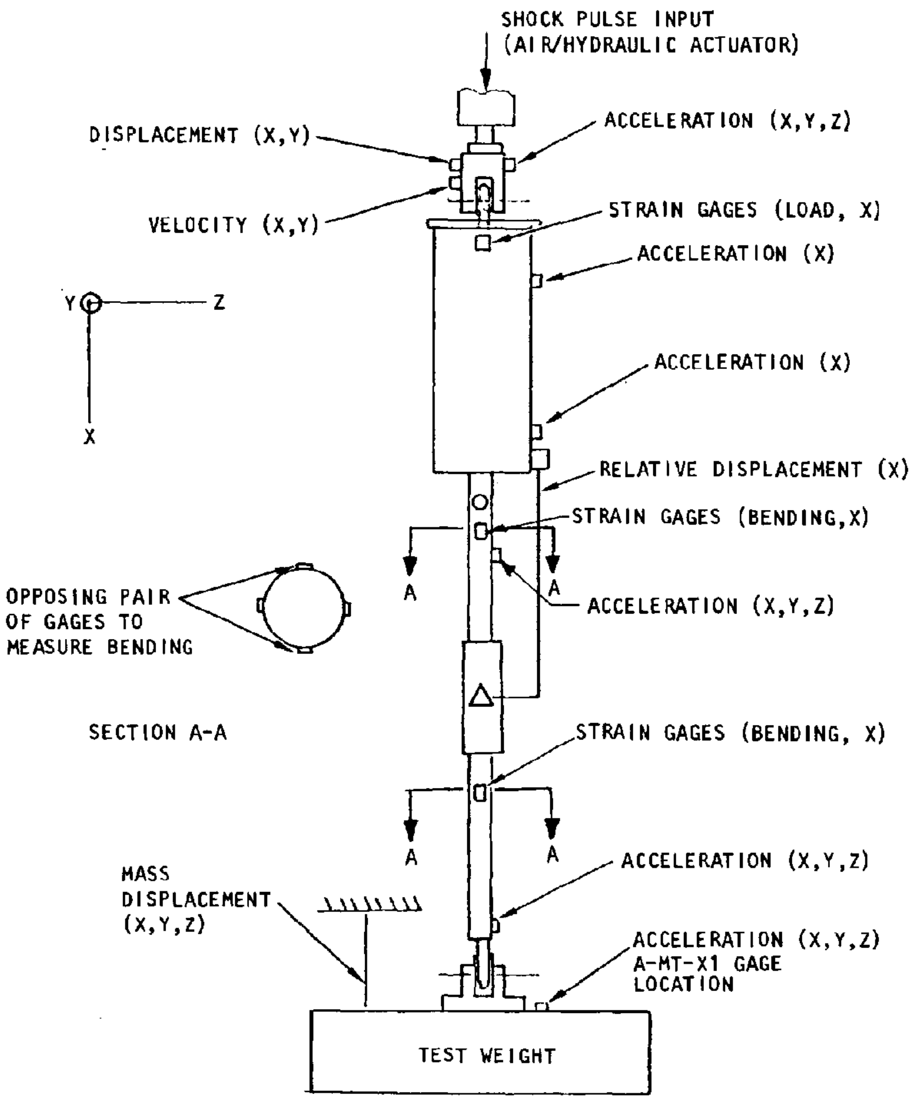


Figure 9-2. Single-Isolator Tests and Instrumentation for a Mechanical Spring (Krek, 1970)

equipment and its support. The indirect method illustrated in figure 9-5 measures the transfer function at a point of practical and physical convenience. Working inversely, i.e., from the motion generator, through the shock-isolation system, to the building, a force-output to force-input ratio is taken. This ratio is identical to the acceleration ratio taken in the opposite direction.

(5) A transfer function may be taken from each location on a platform to each isolator/facility attachment or the transfer function may be summed for all isolators to yield an overall or global transfer function. This latter measurement is convenience if the facility inputs at all isolators are specified to be of equal magnitude and phase. Example transfer functions are given in figures 5-13 to 5-22.

(6) The transfer-function plot of magnitude and phase as a function of frequency must be viewed as a nonparametric representation of the system being measured or used for calculations. That is, the degree of participation of the particular masses, springs, and damping of the system is not identified, although their effects have been measured. To define contributions of such system components requires additional measurements, system modeling, and computer identification routines. Such efforts become desirable if the system is to be modified for improved response by adding stiffness, damping, or special isolators for protection of the equipment.

(7) Transfer-function measurements described by Safford et al. (1977a) uncovered some nonlinearities around isolator/platform resonance frequencies. These nonlinearities were attributed to friction in isolator seals and guides. Transfer-function measurements taken with and without a 1 Hz steady oscillation showed a factor of 1 to 2 difference in magnitude, but both types of measurements converged at approximately 30 hz. For transient input loads with the platform initially at rest, the actual motion will be somewhat less than predicted due to the nonlinearity. This overconservatism is not considered a serious error on the hardness/survivability requirements of the equipment mounted on the platforms.

(8) The motions of shock-isolated platforms are calculated with input from specified facility input motion threats. In the frequency domain, the two functions (input and transfer function) are multiplied, followed by an inverse Fourier transformation to obtain the platform acceleration-time history. In a time-domain computation, the input-time history of the building is convolved with the platform impulse function to obtain the response.

(9) Response motions are determined by the following:

$$\ddot{X}_R(t)(=)\ddot{x}_R(\omega)= \tau(\omega)\ddot{x}_i(\omega) \qquad (9\text{-}2)$$

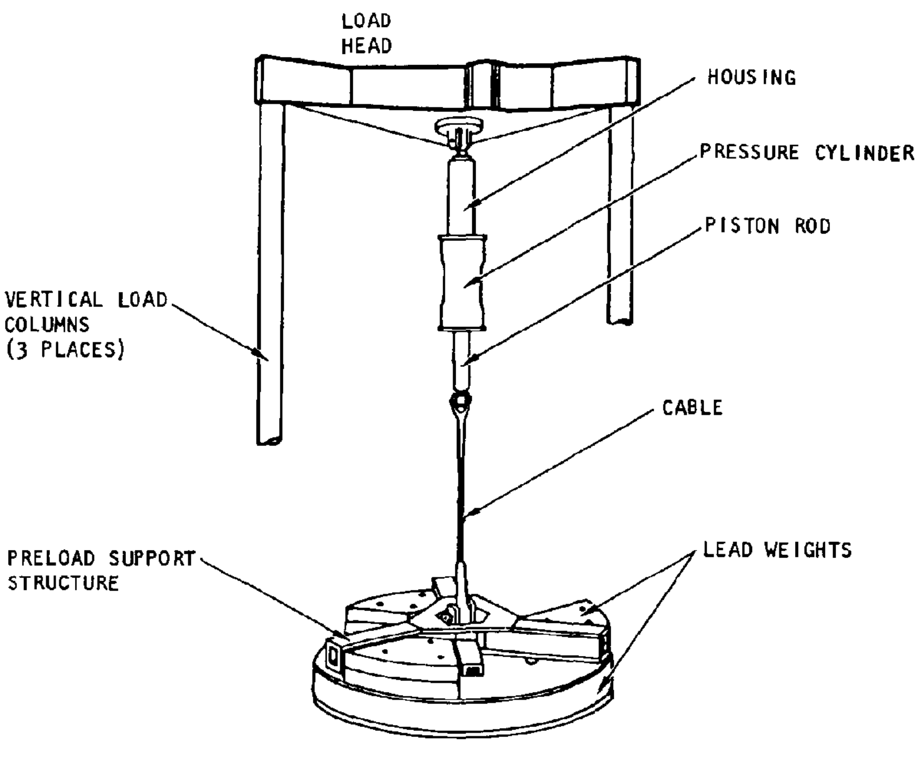


Figure 9-3. Hardmounted Inverted Configuration of Liquid Isolator Vertical Shock Testing (Ashley, 1976)

where

$\ddot{X}_R(t)$ = Acceleration-time history response of platform at reference point

$\ddot{x}_R(\omega)$ = Acceleration-frequency response of platform at reference point

$\ddot{x}_i(\omega)$ = Input building acceleration, Fourier-transformed to frequency domain

$\tau(\omega)$ = Measured overall transfer function of platform

(10) If the facility motion varies for each isolator location, then the response computation becomes the sum of the motions for the individual input locations and associated transfer function paths, which are time delayed in accordance with signal arrival times:

$$\ddot{X}_R(t)(=)\ddot{x}_R(\omega) = \sum_{1}^{n} \tau_n(\omega)\,\ddot{x}_n(\omega)\,e^{-j\tau_n\omega} \qquad (9\text{-}3)$$

where

τ_n = Arrival time of input motion at facility location n

$\ddot{x}_n(\omega)$ = Facility motion at location n

$\tau_n(\omega)$ = Transfer function to platform from location n

(11) The dual of equation 9-2 is the convolution integral from which the identical platform response may also be computed as:

$$\ddot{x}(\omega)(=)\ddot{X}(t) = \int_0^T h(t-\tau)\,\ddot{X}_i(\tau)\,d\tau \qquad (9\text{-}4)$$

where

$\ddot{x}(\omega)$ = Acceleration-frequency response of platform at reference point

$\ddot{X}(t)$ = Acceleration-time history response of platform at reference point

t, T = Time

h(t) = Measured impulse function of platform (transformed from transfer function $\tau(\omega)$)

$\ddot{X}_i(t)$ = Input building acceleration

τ = Time delay function for the convolution operation

(12) Response motion predictions of the control room platform of the Safeguard system is displayed in figures 9-6 through 9-8. Acceleration-time histories for the frequency band 35 Hz to 500 Hz and 0.5 Hz to 500 Hz are included in figures 9-6a and 9-6b. The shock spectrum and Fourier spectrum for this platform response are provided in figures 9-7 and 9-8.

b. Modal survey. The development of digital signal processing hardware and software has progressed to a point where transfer functions (system functions) can be measured easily, rapidly, and with considerable accuracy (Klosterman-Zimmerman, 1975; Richardson-Kniskern, 1976). Using system (transfer) functions, effective modal survey work can be performed on-line. This allows estimation of modal parameters, geometric displays of mode shapes, and checks on measurement accuracy. These developments permit complete mapping of shock isolation systems and extend or permit additional utilization of the acquired transfer function data discussed in the preceding section. With the acquisition of modal response data, performance can be predicted in accordance with the procedures given in chapter 7 for modal analysis.

9-4. Full-scale tests.

a. Partial tests.

(1) Segments of shock-isolated floors equipped with several isolators and some equipment may be tested under severe input loadings. Tests of this type may be required where extremely high rates of input loading are expected to occur. These tests are developmental in character but usually employ floor segments, isolators, and equipments that represent prototype components. The size of the floor segment and the number and weight of isolators used in the tests are governed by test machine capacity.

(2) These partial system tests provide an excellent opportunity for validation of shock isolation systems, particularly under extreme threat conditions. Subsequent data analysis and performance evaluation also permit modifications, upgrading, and improvements to the system.

(3) Partial system tests are illustrated in figures 9-9, 9-10, and 9-11. Figures 9-9 and 9-10 apply to Minuteman upgrade, in which the floor segment is suspended by cable-connected liquid-spring isolators and protected horizontally by foam isolators. Results of these tests can be found in Boyd and Huang, 1977. Figure 9-11 covers a test on a floor segment representing an original Minuteman shock isolation configuration.

(4) Another partial test for equipment "as built" and "as installed" in a protective facility is the drop (twang) test of a shock-isolated platform. These tests are generally benign as regards vulnerability/hardness, but are useful in verifying the stroking characteristics of the isolators, the characteristics of damping, and the coupling of modes. Additional information may also be obtained about rattlespace adequacy, rattlespace incursions (cables and piping), and the effect of electrical cables and piping loops connected to the platform.

(5) Current practice is to elevate the platform to a specified stroke displacement and then suddenly release the elevating mechanism. This test technique is shown in figure 9-12. Another type of lifting device used to twang-test Minuteman floors is sketched in figure 9-13. The twang devices shown in figure 9-12 are part of the inventory at the Weapons Effects Laboratory, Waterways Experiment Station.

(6) In some early twang tests, the platforms were pulled down with mechanical cables. However, the quick-release devices on the cables generated such a high transient vibration in the platform that much of the response data was obscured.

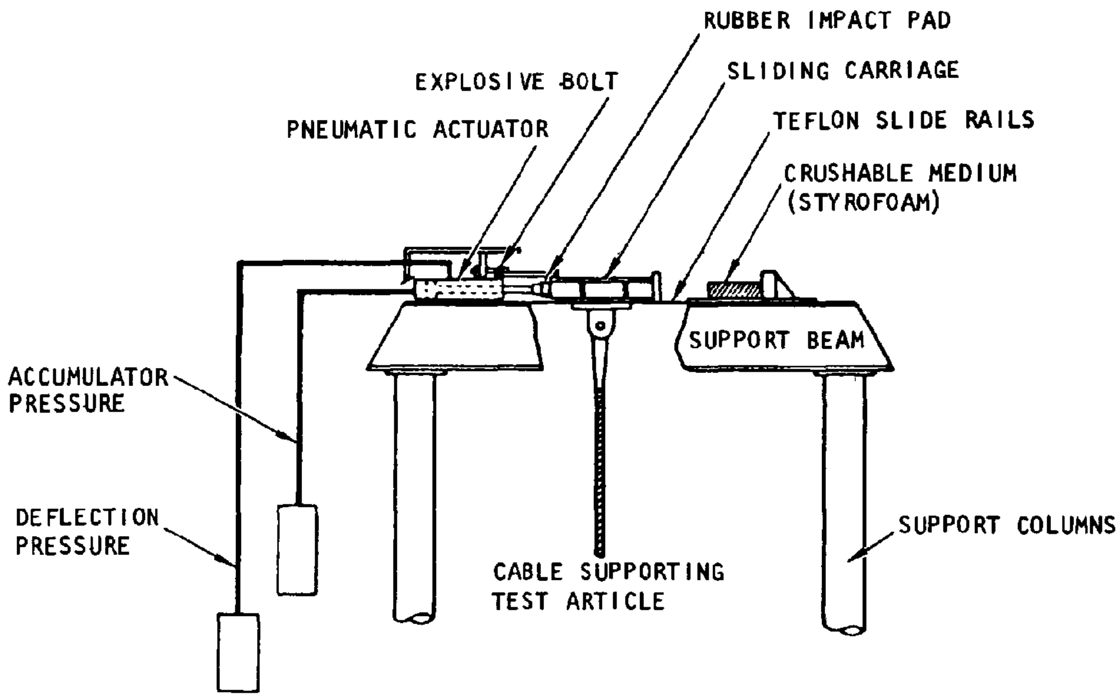


Figure 9-4. Horizontal Shock Testing of Cable-Suspended Liquid Isolator (Ashley, 1976)

b. *Quality control tests.*

(1) Quality control vibration tests should be performed on installed shock-isolated platforms just before the system is turned over to the operational command. In this procedure, one or more vibrators (1000 to 2000 lb output force) are mounted directly on the platform, and tests are conducted with equipment operating. Vibration input can be shaped into random or sweeping sine waves over the frequencies of interest. Amplitudes of vibration can range up to the predicted response levels of the platform to an attack. Test durations may approximate the expected threat durations.

(2) These quality control (QC) tests are expected to uncover defects in workmanship, weakness in material, and errors in installation and hook-up. Periodic QC tests should also be conducted during the life of the facility to guard against deficiencies arising from the maintenance as well as from removal, replacement, and upgrade of equipment.

(3) Some years ago a simple impact test was employed for testing operational Minuteman equipment platforms. The device used is shown in figure 9-14.

c. *Testing of protective facilities.* Testing of full-size protective facilities or models of facilities containing

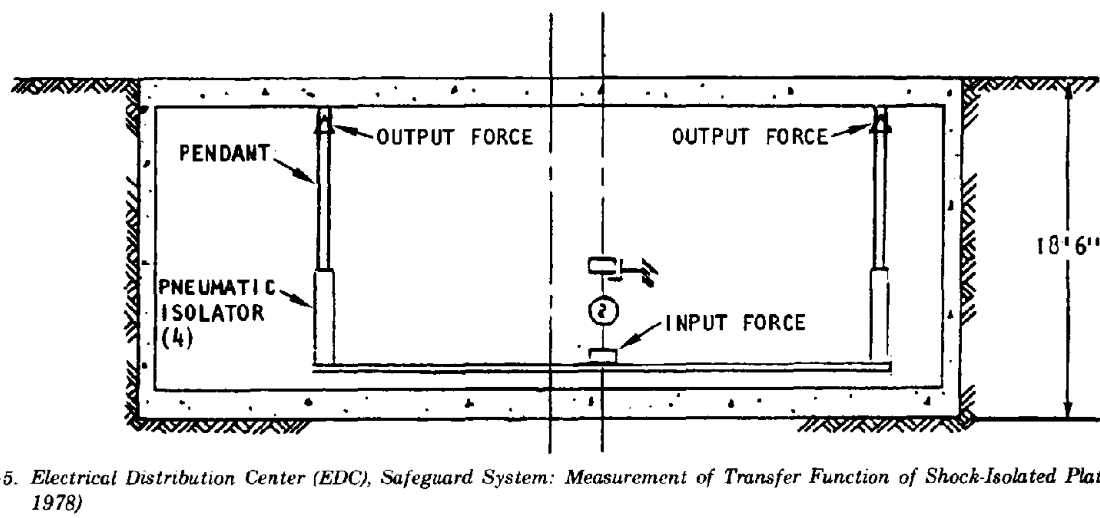
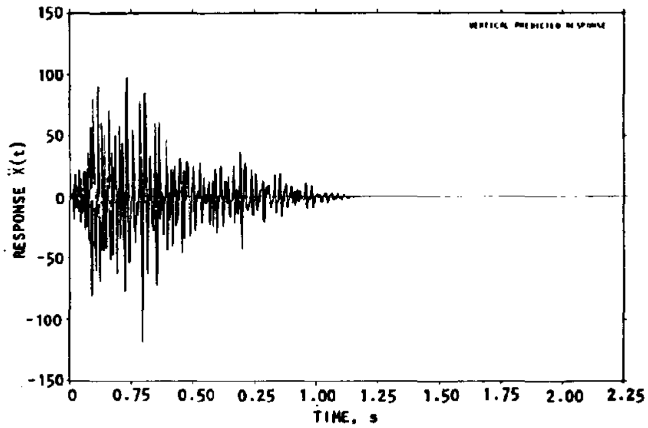
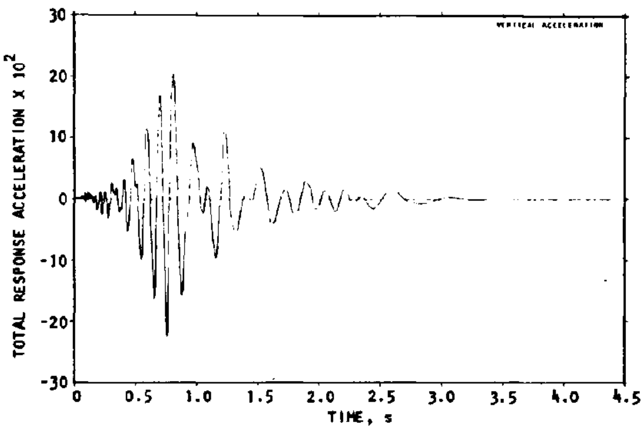


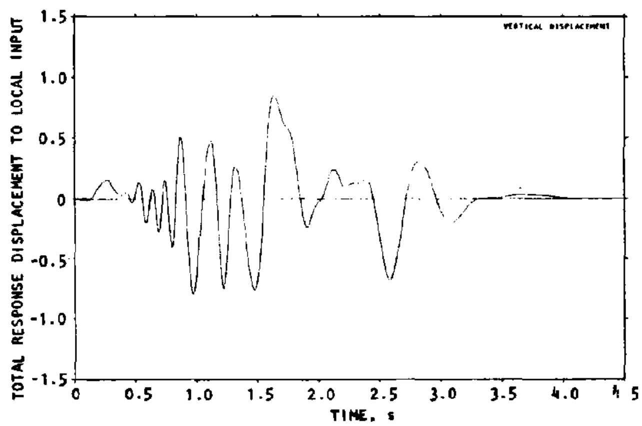
Figure 9-5. Electrical Distribution Center (EDC), Safeguard System: Measurement of Transfer Function of Shock-Isolated Platform (AA, 1978)



(a) Predicted acceleration time history response, using the computed global transfer function convolved with overall input (frequency range, 35 to 500 Hz)



(b) Acceleration time history for overall input (frequency range, 0.5 to 500 Hz)



(c) Displacement time history for local input (frequency range, 0.5 to 500 Hz)

Figure 9-6. Response Motion Prediction of Control Room (Safford-Walker, 1975; Safford et al., 1977)

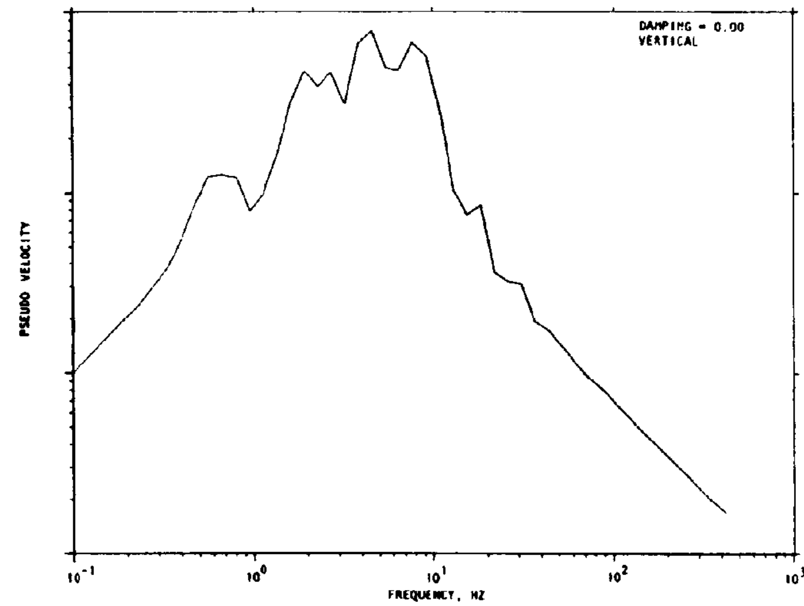


Figure 9-7. Control Room Platform: Vertical-Response Shock Spectrum for Overall Input (Frequency Range 0.5 to 500 Hz) (Safford-Walker, 1975; Safford et al., 1977)

shock-isolation systems may be conducted by several techniques. These testing techniques include the following:

—High-explosive simulation test (HEST)

—Berm-loaded explosive simulation test (BLEST) and HEST-BLEST

—Direct-induced high-explosive simulation test (DIHEST) and HEST-DIHEST.

—High-explosive contact surface burst (HEFIELD TEST)

—Underground nuclear tamped burst and tunnel test

A description and discussion of each of these tests is provided in TM 5-858-6, *Hardness Verification*, chapter 4, "Verification Requirements and Experimental Methods."

d. Shock isolation system tests.

(1) A shock-isolated platform of the Minuteman launch-control room was tested on the shock-simulator shown in figure 9-15. Velocity pulses were applied to simulate ground shock conditions. This shock simulator was located at the Air Force Weapons Laboratory (AFWL), Kirtland AFB, New Mexico, and has since been scrapped.

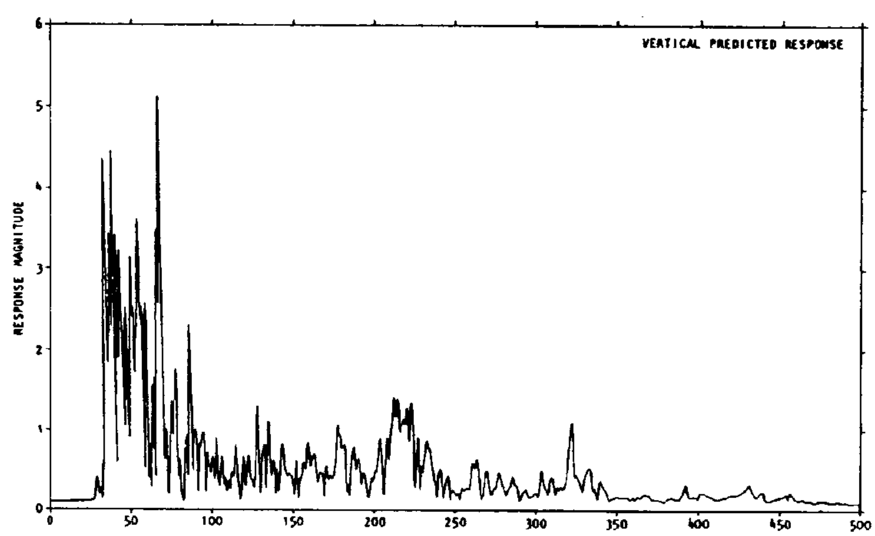


Figure 9-8. Control Room Platform: Predicted Acceleration Fourier Response Magnitude, Using the Computed Global Transfer Function Convolved with Overall Input (Frequency Range 35 to 500 Hz) (Safford Walker, 1975; Safford et al., 1977)

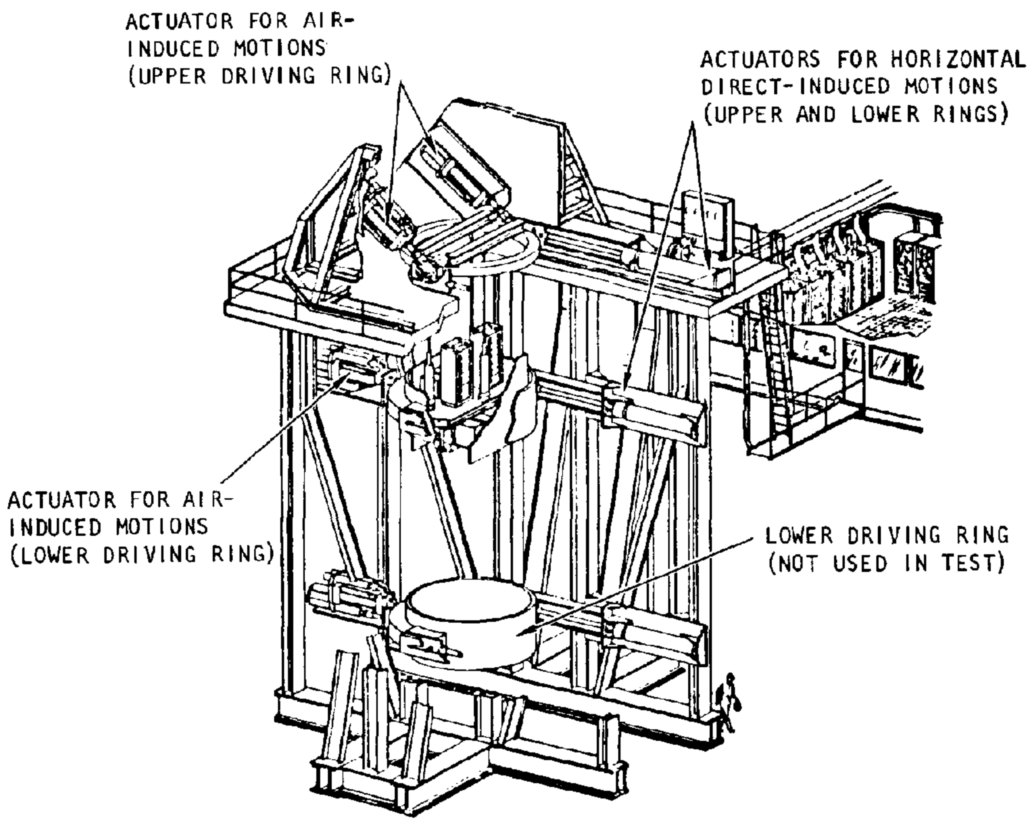


Figure 9-9. Dynamic Test Setup with Floor Segment in Place (Gustafson, 1976)

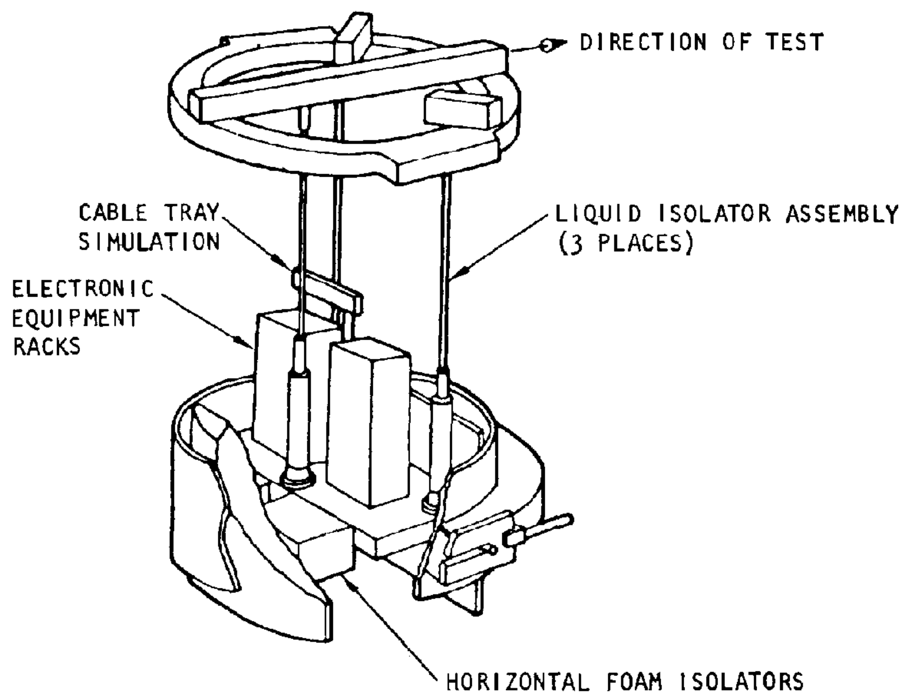


Figure 9-10. Schematic of Floor-Segment Test Specimen (Gustafson, 1976)

(2) The preceding test concept was developed further by the Boeing Co. for the U.S. Air Force Space and Missile Systems Organization (SAMSO) for higher level excitation having both positive and return strokes. This new system was designed to test Minuteman shock-isolated equipment platforms and the Minuteman missile/missile-isolation system.

(3) Performance characteristics of these new actuators are listed in table 9-2. Shock spectra and waveform performance requirements are presented in figures 9-16 and 9-17. Schematics of the actuator are detailed in figures 9-18 and 9-19. Test results for both positive and return motion are presented in figures 9-20 and 9-21.

(4) Tests for the missile/missile-isolation system are shown in figure 9-22. This system is presently located at the Boeing Company, Seattle, Washington. Four actuators were used for testing the shock-isolated floor, as depicted in figure 9-23. This system is presently located at Vandenburg Air Force Base, California. Numerous tests of shock-isolated platforms have been conducted. Among these tests were several where the isolators were overdriven to impact and subsequently failed.

e. Pulse simulation tests.

(1) In-place testing of equipment mounted on shock-isolated platforms yields information for the assessment of hardness and vulnerability to a specified nuclear threat of ground shock and airblast. One advantage is that in-place tests can be performed, in which individual items of equipment are electrically interconnected and are also connected into a major system such as the power or signal distribution network. Function monitoring to detect system failures, malfunctions, or marginal performance is performed with equipment operating before, during, and after in-place tests. Results of in-place tests provide for evaluation of both equipment performance and its impact on total weapon-systems performance.

(2) The in-place tests require estimates or predictions of shock-isolated platform motions that would be caused by specified nuclear environments. These predictions are made by convolving the specified input facility environments with the measured platform transfer functions. This procedure is covered in paragraph 9-3. Motion may also be predicted by the computational techniques covered in chapter 7, particularly those using finite element procedures.

(3) The need for a mechanical force-pulse generator arose in connection with studies on how to effectively simulate, by physical test, shock transients on shock-isolated equipment located in Safeguard protective structures. There was a restriction that force generators could not be interposed between the shock isolators and the building. The platform environments represent the building motions generated by nuclear

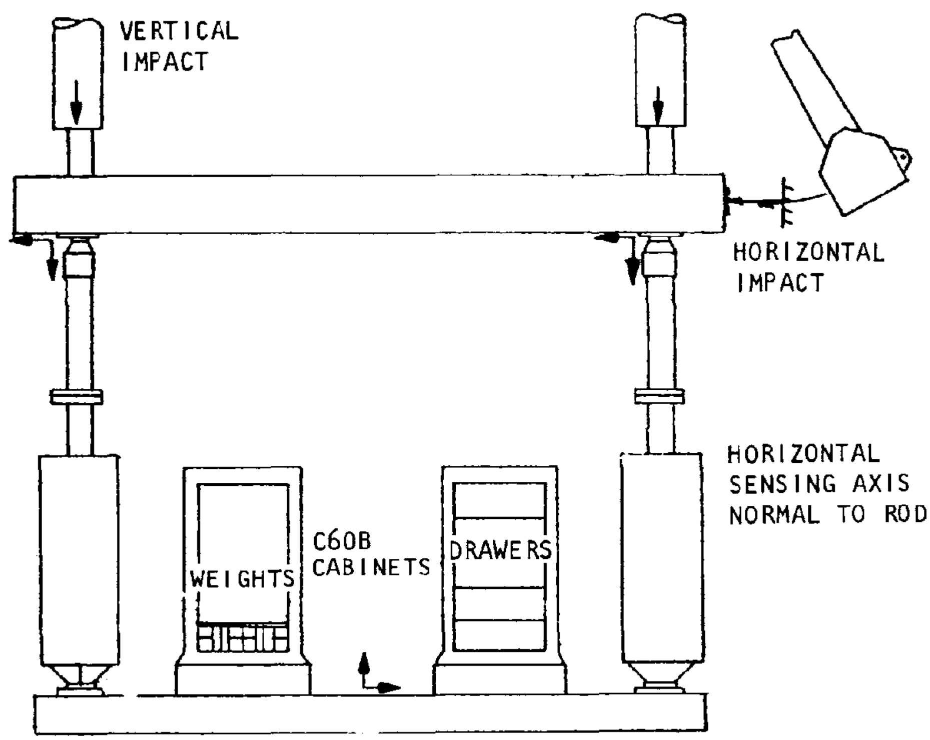


Figure 9-11. Dual Isolator Beam Segment Test (Boeing, 1977)

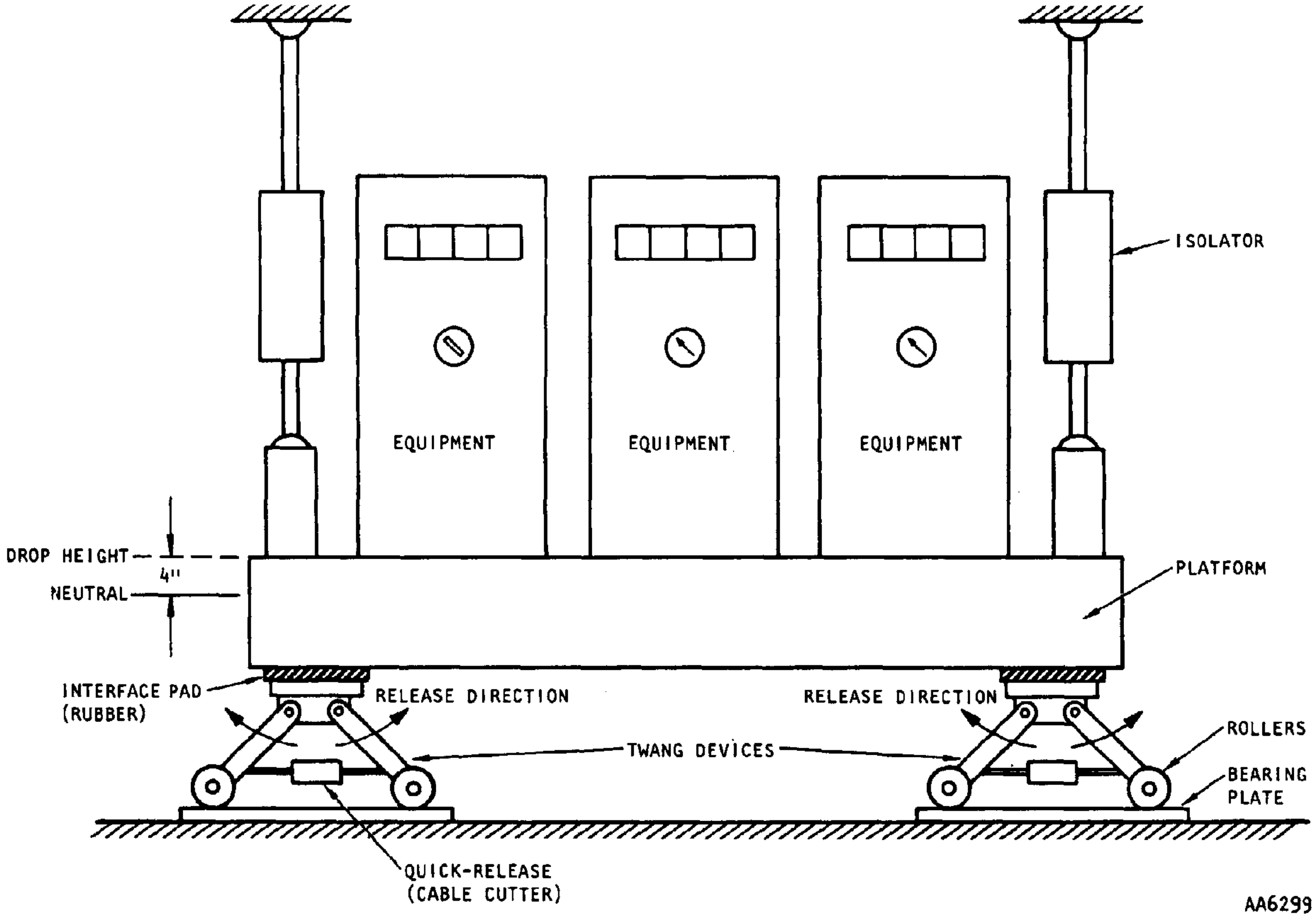


Figure 9-12. Sketch of Twang Test Technique for 4-in Drop Test of Shock-Isolated Platforms (Safford-Walker, 1975)

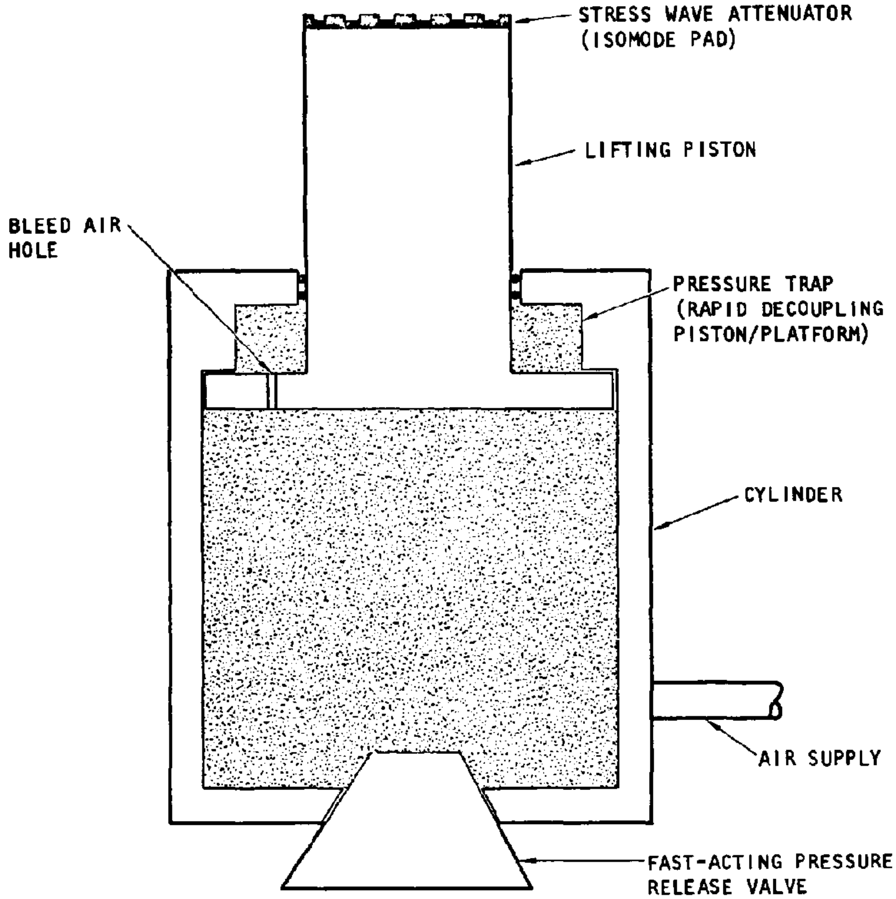


Figure 9-13. Twang Device to Lift Platforms and to Release Platforms Rapidly from Offset Position (Wyle Corp.)

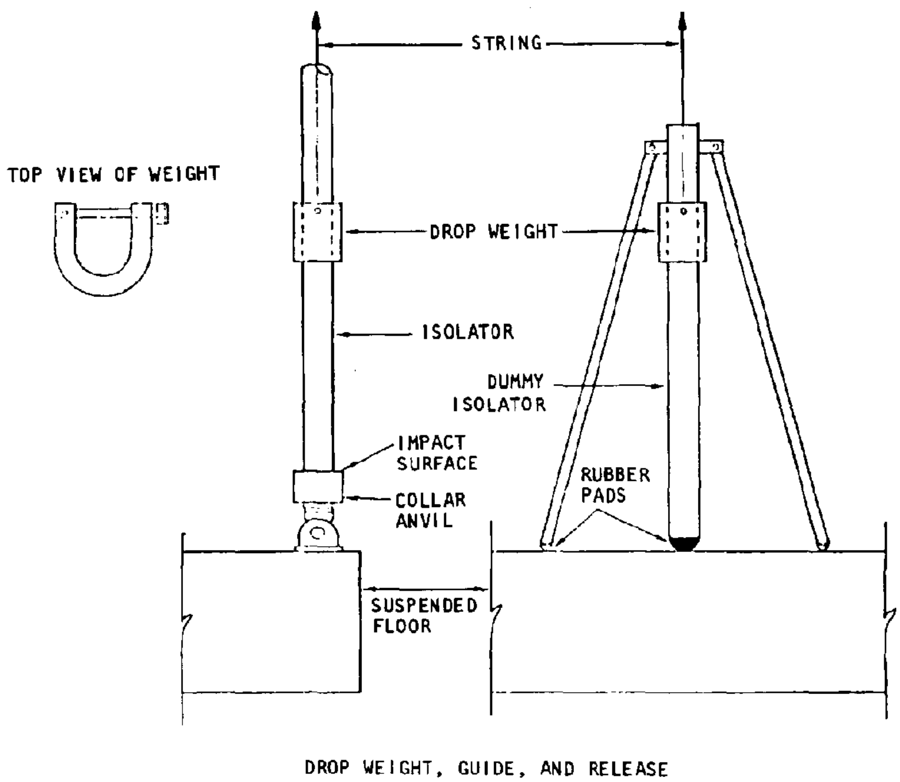
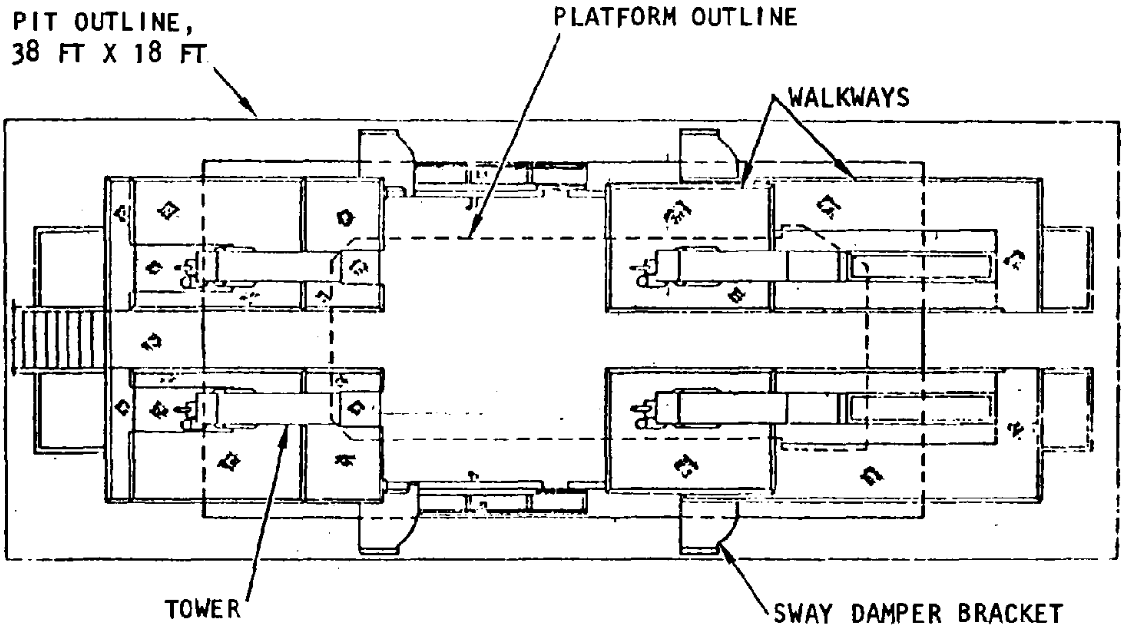
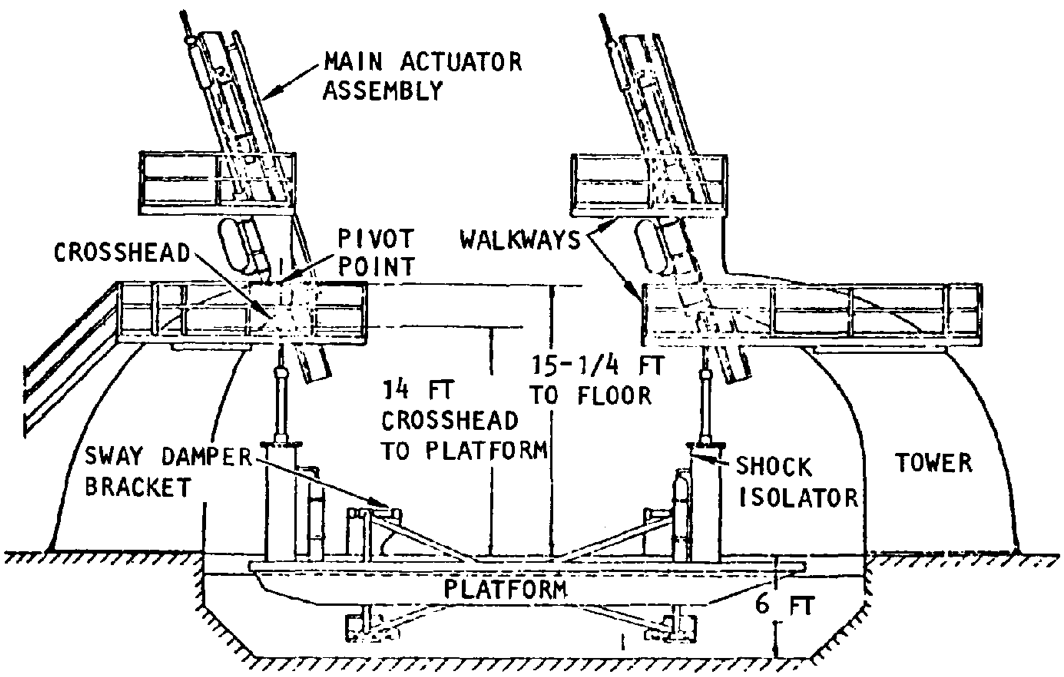


Figure 9-14. Production Environmental Tests on Isolated Launch-Control Equipment Platforms of Minuteman (TRW)



(a) Plan view of shock simulator



(b) Elevation view of shock simulator

Figure 9-15. Launch Control Center Isolated-Platform Tests (Parsons, 1962)

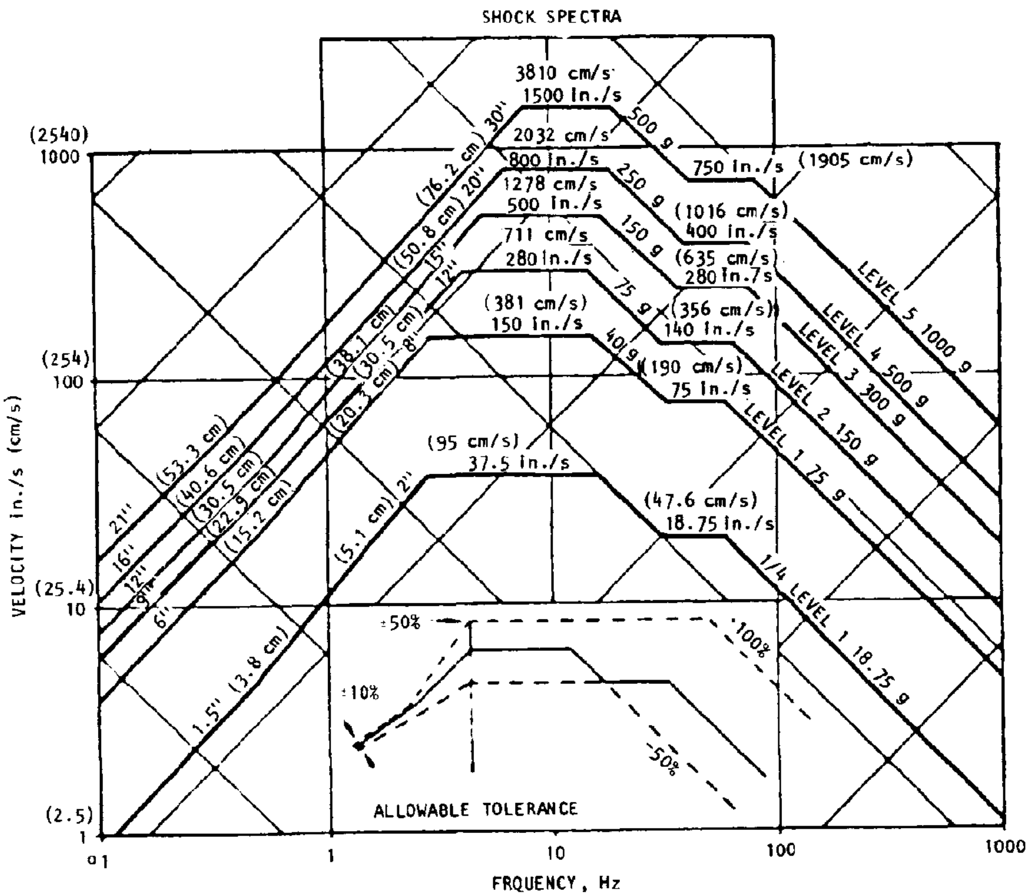


Figure 9-16. Primary Actuator Design Shock Levels (Burwell, 1976)

TABLE 9-2. PERFORMANCE CHARACTERISTICS OF BOEING SHOCK ACTUATOR

	Parameter	Maximum	Minimum
Test Article	Driven Weight	16,200 lb (7348 kg)	1,900 lb (862 kg)
	Static Load	60,000 lb (27,200 kg)	0.0 lb (0.0 kg)
Measured Peak Values	Acceleration (Rigid Body)	800 g	17 g
	Velocity Positive Return	 1040 in./s (2642 cm/s) 190 in./s (483 cm/s)	 40 in./s (102 cm/s) 13 in./s (33 cm/s)
	Displacement Positive Return	 36 in. (91.4 cm) 11.5 in. (29.2 cm)	 1.5 in. (3.8 cm) 0.8 in. (2.0 cm)

U.S. Army Corps of Engineers

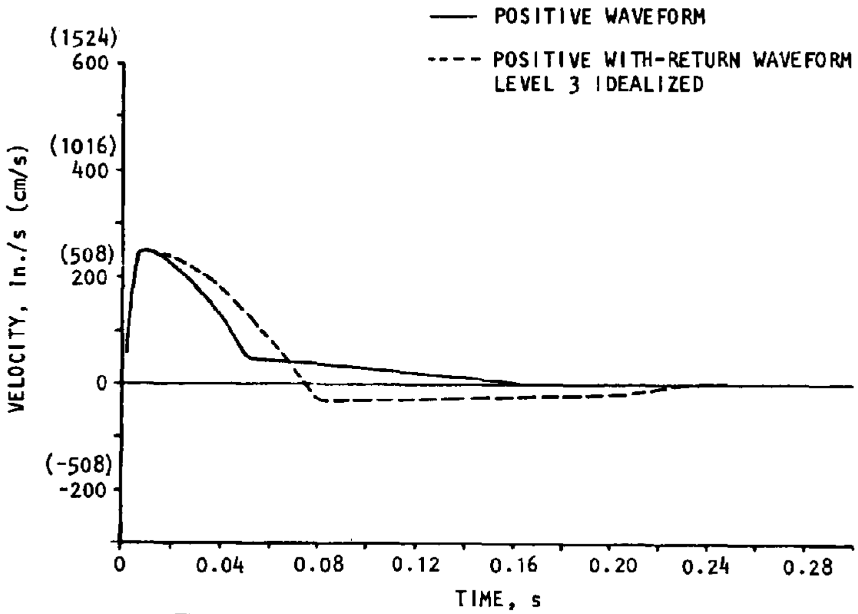


Figure 9-17. Actuator Waveform Types (Burwell, 1976)

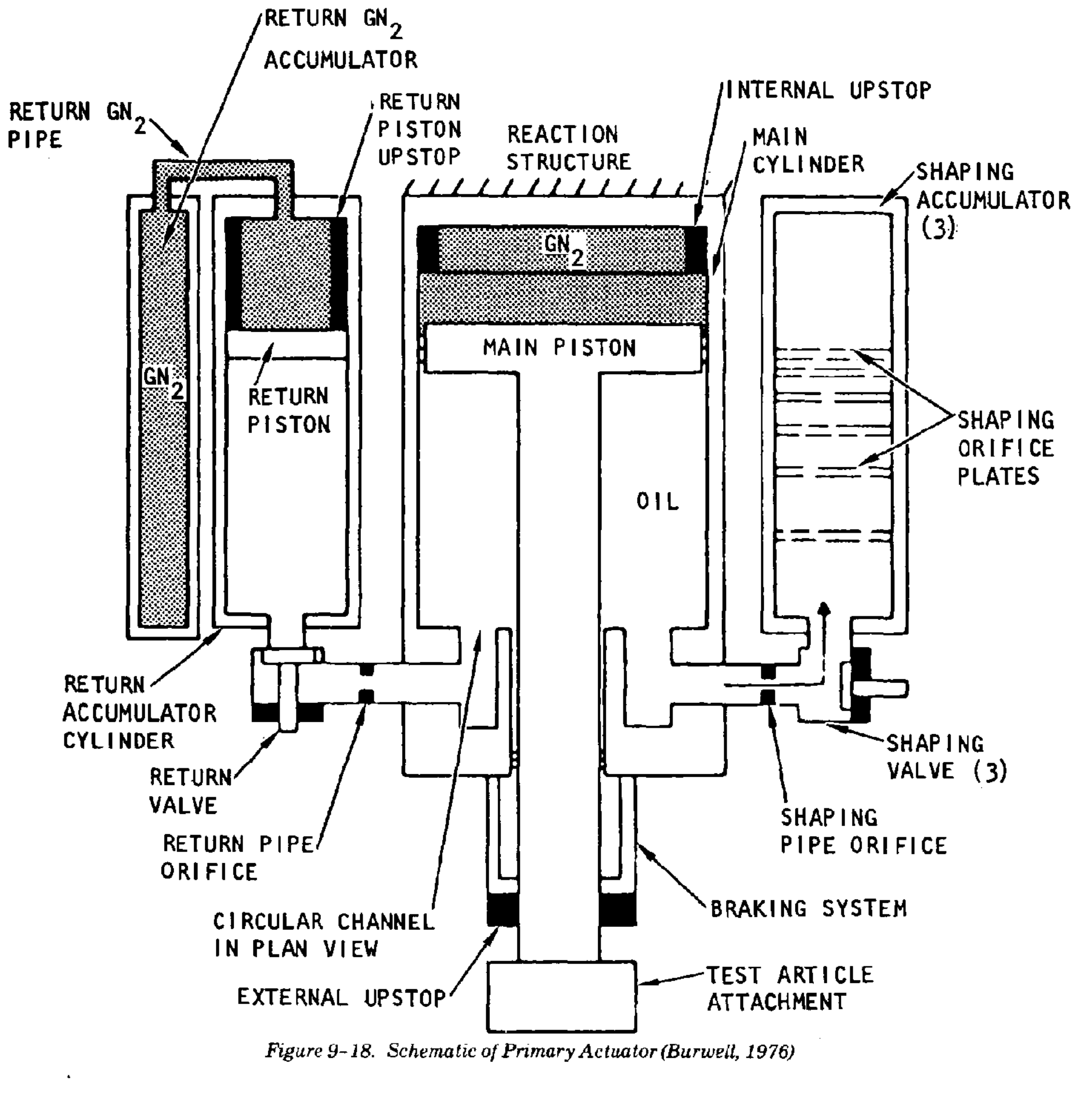
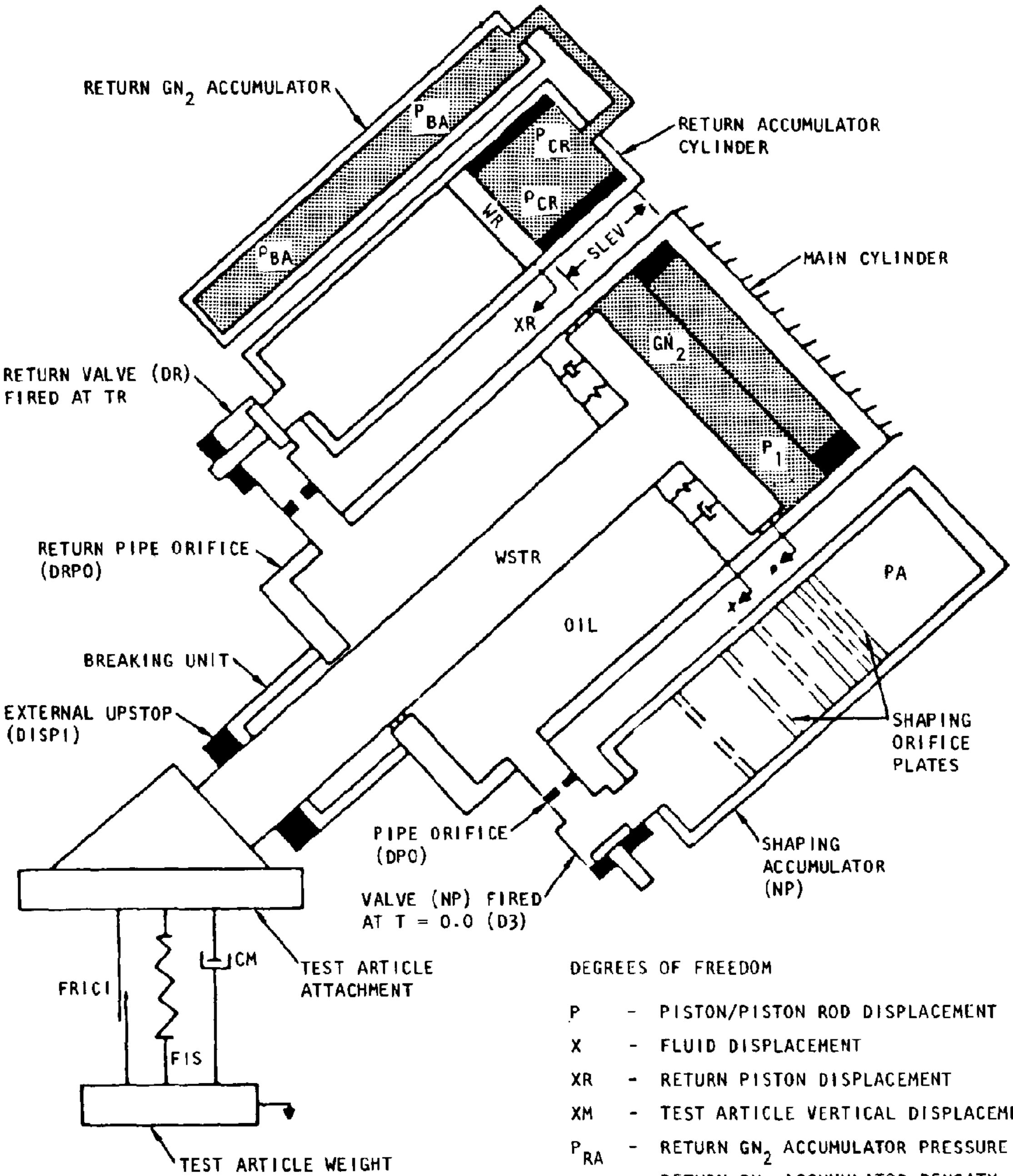


Figure 9-18. Schematic of Primary Actuator (Burwell, 1976)



DEGREES OF FREEDOM

- P - PISTON/PISTON ROD DISPLACEMENT
- X - FLUID DISPLACEMENT
- XR - RETURN PISTON DISPLACEMENT
- XM - TEST ARTICLE VERTICAL DISPLACEMENT
- P_{RA} - RETURN GN_2 ACCUMULATOR PRESSURE
- ρ_{RA} - RETURN GN_2 ACCUMULATOR DENSITY
- P_{CR} - RETURN ACCUMULATOR CYLINDER GN_2 PRESSURE
- ρ_{CR} - RETURN ACCUMULATOR CYLINDER GN_2 DENSITY

Figure 9-19. Actuator Inclined for Vectored Input: Schematic of Idealized Actuator (Burwell, 1976)

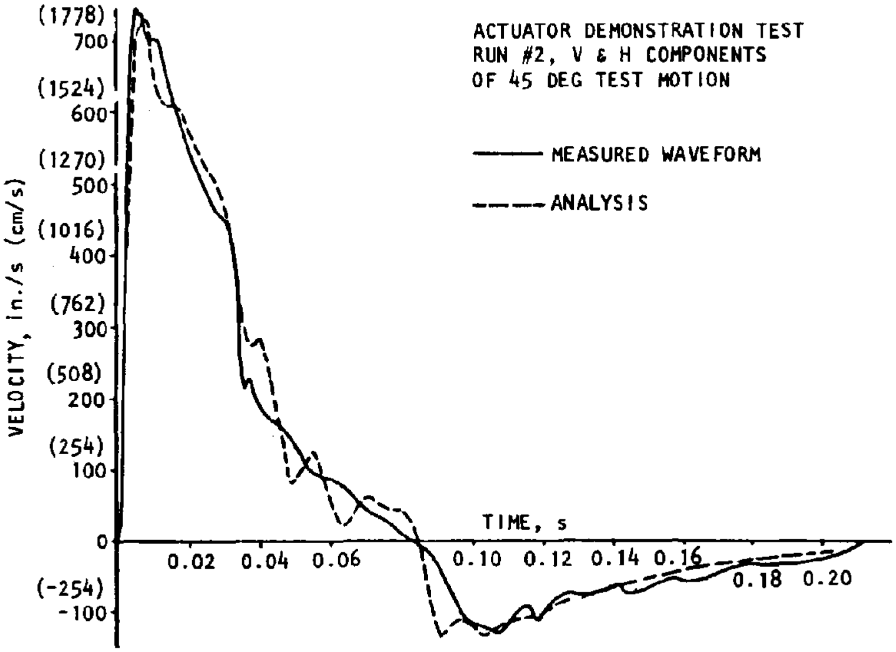


Figure 9-20. Test/Analysis Waveform Comparison, Positive-with-Return, Vertical and Horizontal Components of Maximum Velocity Capability (Burwell, 1976)

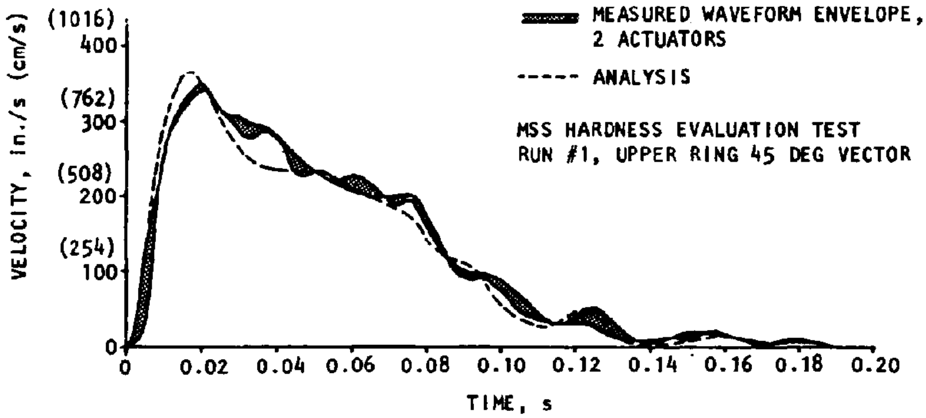


Figure 9-21. Test/Analysis Waveform Comparison, Positive Only, Missile Suspension System Test (Burwell, 1976)

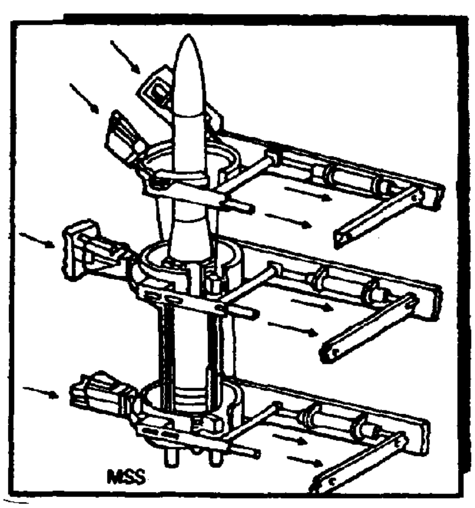


Figure 9-22. Missile-Isolation System and Missile, Mounted for Test in Shock Simulator (Boeing, 1977)

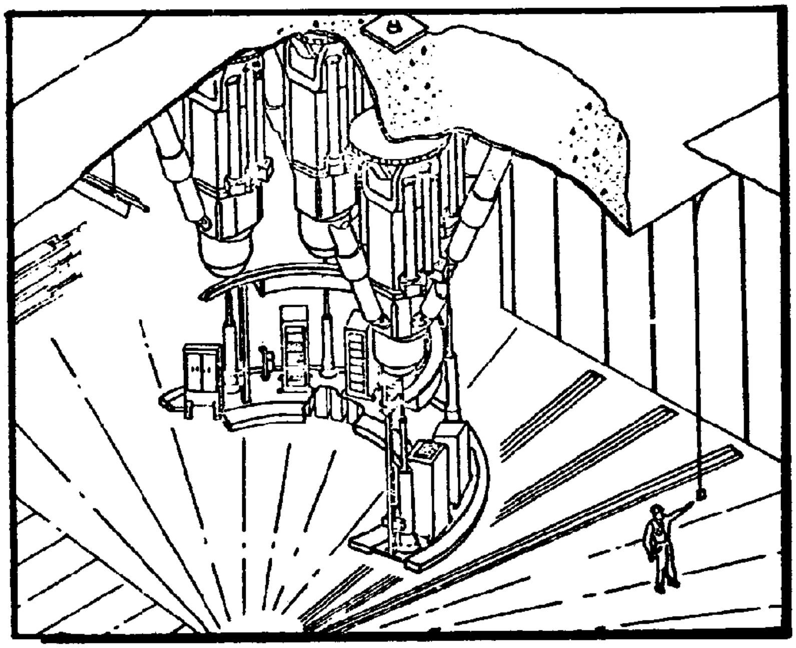


Figure 9-23. Shock-Isolated Platform Mounted for Shock Test (Four Actuators) (Boeing, 1977)

attack and transmitted into the equipment through the isolation system. The weight of the shock-isolated platforms and mounted equipment to be tested in place ranged from 1400 lb to 284,000 lb.

(4) The concept for a mechanical pulse generator simply reverses a device for energy absorption to obtain force output of the desired profiles. By drawing a metal bar or mandrel through a cutting tool (or vice versa) with suitable motive power (air pressure, hydraulic pressure, explosive force, electrical, mechanical), a series or a set of force time histories may be generated (Safford-Masri, 1974). Reaction at the attached points of the device transmits a force output to the structure under test. Figure 9-24 illustrates the device using motive power supplied by the stored energy in a pneumatic cylinder. Amplitude, duration, and shape of the pulse time histories are controlled by the relative velocity between the cutting tool and the metal projection on the mandrel, and by the shape of the metal projection on the mandrel. In profile, the mandrel projection shapes show a very wide range, such as saw tooth, sine, versine, hyperbolic, and rectangular.

(5) Forces may be generated singly and in series for one cutting head or in parallel with multiple cutting heads. A single large cutting tool may be used, if economical. The metal may be groove cut, i.e., the work is wider than the tool width; or single cut, i.e., the tool is wider than the work. The cutting tool may have a wide variety of shapes to suit any force-output requirement.

(6) Large forces may be generated from this device. The force required to cut metal is largely independent of rate (velocity) and is a function of the volume of chips cut (depth, width, and length of cut) and of the specific energy of cutting a material. The energy absorbed in metal cutting, as given by COE (1974), is approximately:

$$F\ell = \ell w t \mu \qquad (9\text{-}5)$$

where

F = Force of cutting, lb

ℓ = Length of cut, in.

w = Width of cut, in.

t = Depth of cut, in.

μ = Specific energy of cutting, in.-lb/in.^{3}

μ_s = 3×10^5 in.-lb/in.^{3}, mild steel

μ_{Al} = 1.5×10^5 in.-lb/in.^{3}, aluminum

A load cell or strain gage may be incorporated in series with the device to provide a force-time history readout as the device is operated.

(7) Four pulse generators were used in the Safeguard system for each test and were attached near the four corners of each platform. The schematic of the pulse generator and power-actuation system is given in figure 9-25. Power to the pulse generators was provided by a hydraulic cylinder having volumetric compensators in series. All four units were connected in series by metal piping. A pneumatic-hydraulic accumulator provided stored energy to drive the generators at line pressures to 1000 psi. Pulse-initiation timing was accounted for by pre-positioning each cutter/mandrel and performing calibration runs of the entire pulse system for each platform configuration. The pulse simulation system has been improved for operational and in-place testing on several other projects. The system is maintained by and is part of the inven-

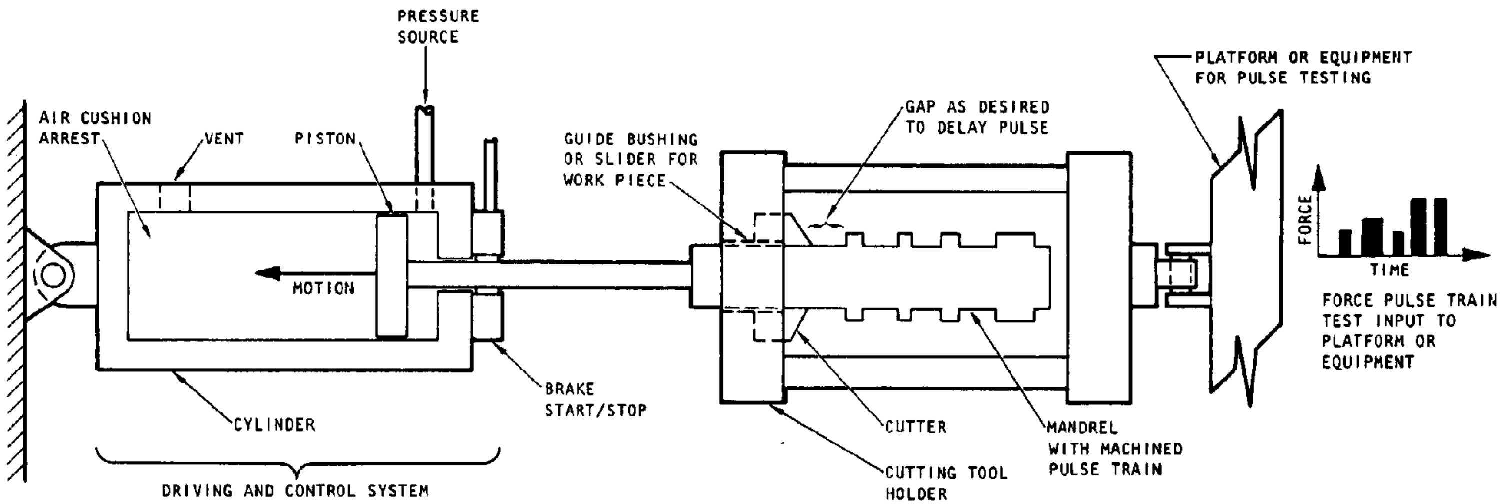


Figure 9-24. Pulse-Forming Device with Driving and Control System (Safford-Masri, 1974).

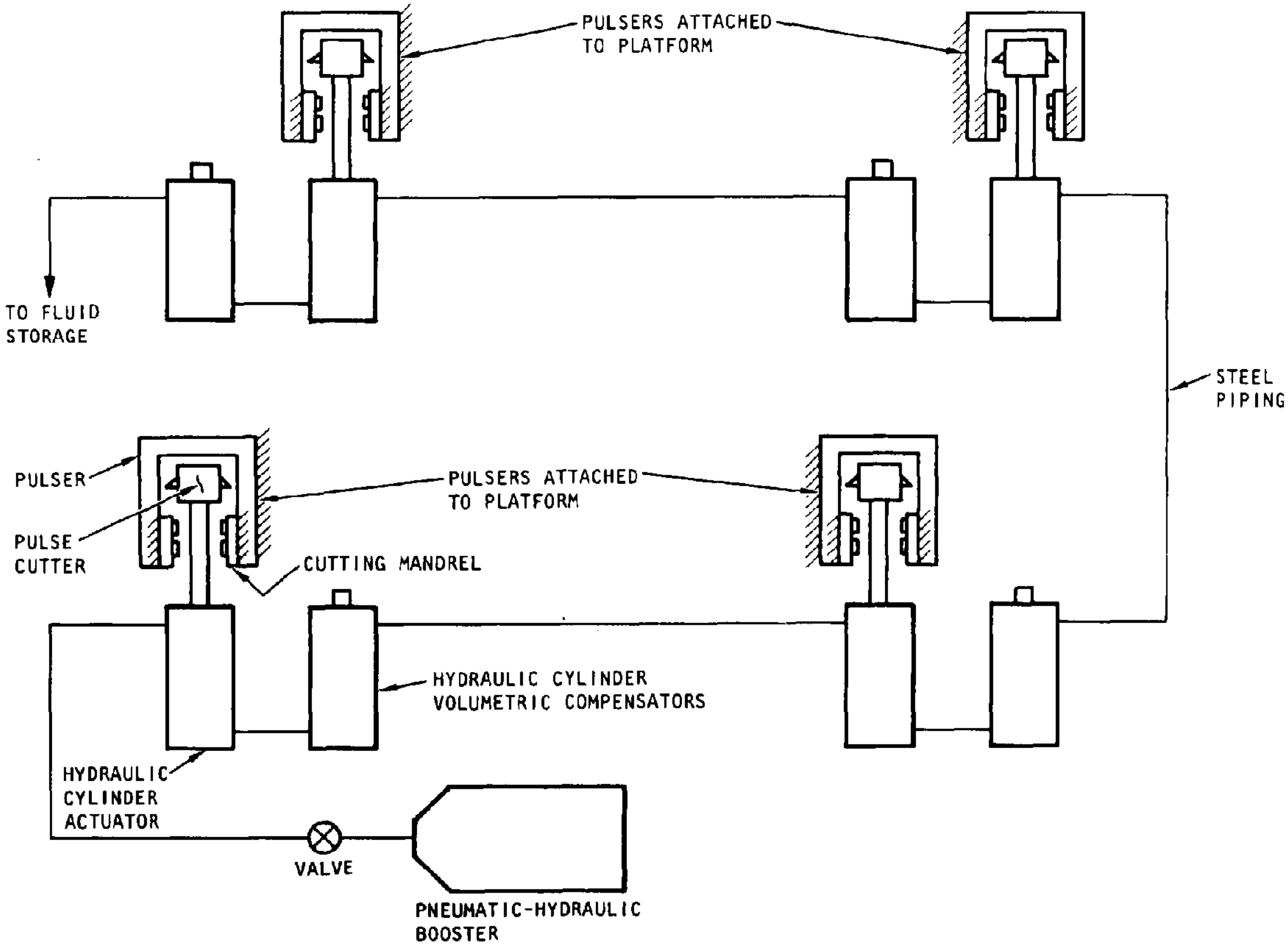


Figure 9-25. Schematic of Pulse-Generator and Power-Actuation System (Safford-Walker, 1975)

tory of the Weapon Effects Laboratory, U.S. Army Waterways Experiment Station.

(8) For the in-place pulse-simulation process, a discrete number of pulses appears quite different from a continuous excitation signal. It is necessary, therefore, to select the pulses in such a way that the resulting vibration of the platform matches as closely as possible the response (i.e., displacement, velocity, or acceleration) produced in the platform by the continuous excitation resulting from the building motion (nuclear threat input). The accuracy of simulation is determined by an appropriate error criterion, as shown in figure 9-26.

(9) It is important to note that the method of figure 9-26 requires that the response to the nuclear threat input be known, which would generally not be true in practice. If this response is unknown, the approach used requires that (1) a mathematical model or the measured transfer functions of the system under study be available, and (2) the inputs of interest (nuclear threat) be given. Under these conditions the "criterion response," i.e., predicted response to nuclear threat input, can be calculated and used to obtain the pulse train for the simulation test. The criterion (predicted) response of platforms was computed from the input (specified) threats and the measured transfer functions for the Safeguard project (Safford-Walker, 1975; Safford et al., 1977a)

(10) In general, the response time history of an article under simulated test should show a reasonable approximation to the expected environment for meaningful hardness/vulnerability evaluation (chap. 4). The following discussion is concerned with the application of a computer algorithm for the optimum selection of a finite number of pulse heights, pulse durations, and onset times to accomplish the simulation.

(11) The basic criterion used for simulation accuracy is the integral squared error between the criterion and pulse-simulated platform response. The error function is evaluated at a sufficient number of points within the multiple-degree-of-freedom platform to characterize the platform as completely as possible. With the error criterion given, the pulse occurrence times, pulse widths, and pulse amplitudes are selected by a systematic-search algorithm such that the error is minimized.

(12) The optimization method used is a simple relaxation algorithm (Aoki, 1971) in which each characteristic of the pulse train is changed repeatedly within certain bounds established by system characteristics and design limitations. It was found that a convenient way of constructing the optimum pulse train is to determine the best single-pulse characteristics (initiation time, amplitude, amplitude, and duration) within a specific segment of time. By then adding the cumulative effects of these segments, the complete system response is found for any prescribed time interval over which the criterion response is to be matched.

(13) Because of the nature of the optimization procedure, repetitive evaluation (thousands of repetitions) of system response to a given pulse was mandatory. It was necessary to develop an efficient method for evaluating the system response during the period $t_i - t$, given the nonzero initial condition of the system at time t_i.

(14) The foregoing method was applied to four platforms. In order to use the simulation method in conjunction with the optimization procedure discussed above, the following steps were performed:

(a) The impulse responses for each platform were determined for each pulse location. This was accomplished by converting measured transfer impedance functions in the frequency domain to transfer impedence impulse functions in the time domain. Typical functions for each platform are shown in figure 9-27.

(b) Using the optimization algorithm, the criterion platform response was simulated by pulse trains convolved with the above impulse function. Typical computed pulse trains are shown in figure 9-28.

(15) Using the pulse profiles specified above, piston velocities of the hydraulic system (fig. 9-25) were established and mandrels of the pulses were machined. Measured pulse trains from in-place tests are also shown in figure 9-28. Typical acceleration-time histories obtained are displayed in figure 9-29. These figures are a three-way comparison of (1) criterion (predicted) response to nuclear threat, (2) pulse-simulated response, and (3) actual response of platforms to in-place pulse tests. The same three-way comparison for each platform is also shown in shock spectrum format in figure 9-30.

9-5. Shock spectra.

a. Prediction of higher frequency response of shock-isolated platforms.

(1) Prediction of the frequency response of shock-isolated platforms prior to construction and installation in a protective facility is a critical requirement. These platforms response predictions are used to establish the qualification, production, and fragility environmental tests for procurement of equipment that will be subsequently mounted on these shock-isolated platforms.

(2) Chapter 7 covers dynamic analysis, particularly finite element procedures that may be used for prediction of platform response. However, accurate prediction is extremely difficult due to the complexity of the equipment/platform/isolators and due to the high frequency involved (500 Hz). In some circumstances, frequencies of 2000 Hz and above may be of concern.

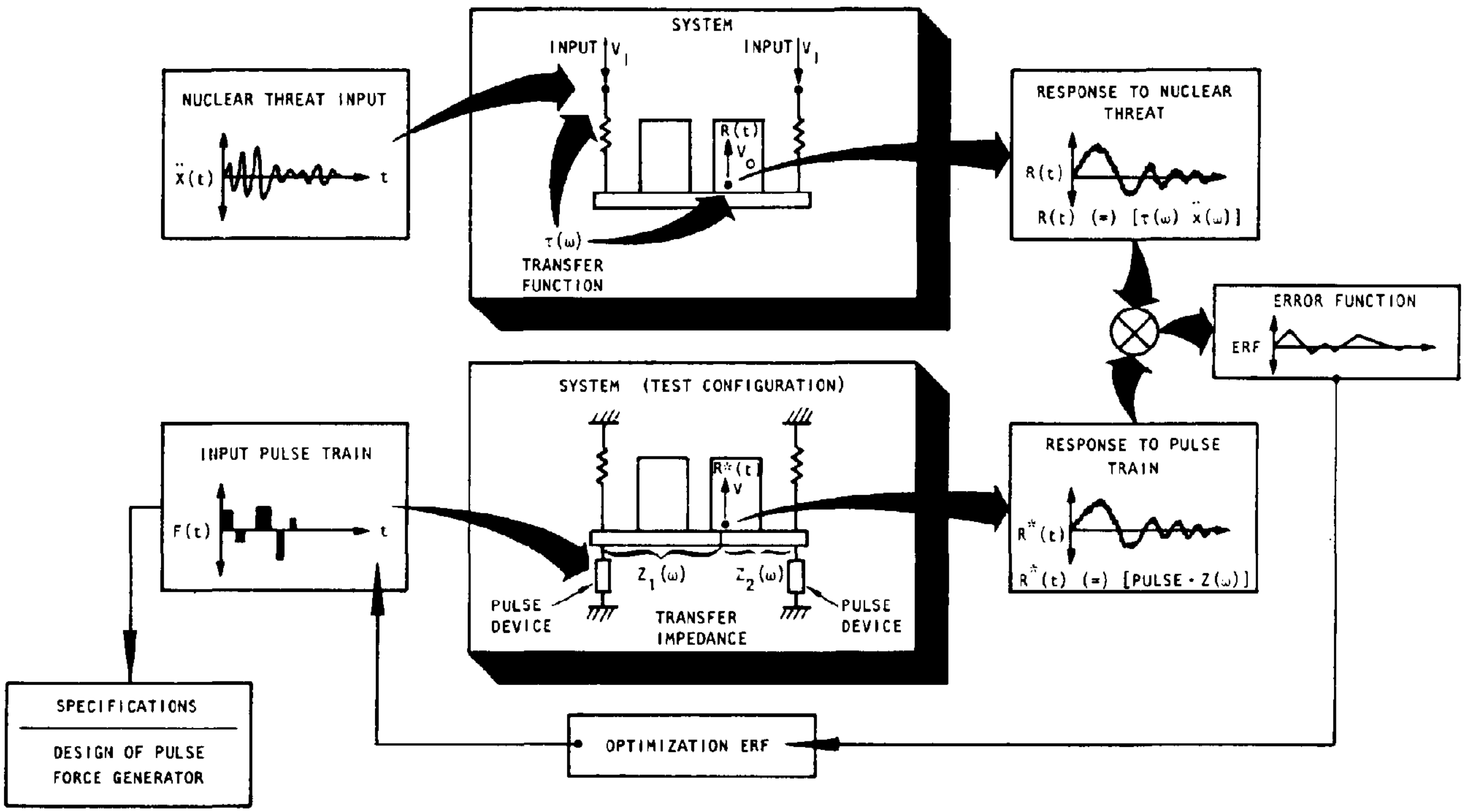


Figure 9-26. Data Flow to Simulate Predicted Response of Platform to a Nuclear Threat (Safford-Walker, 1975)

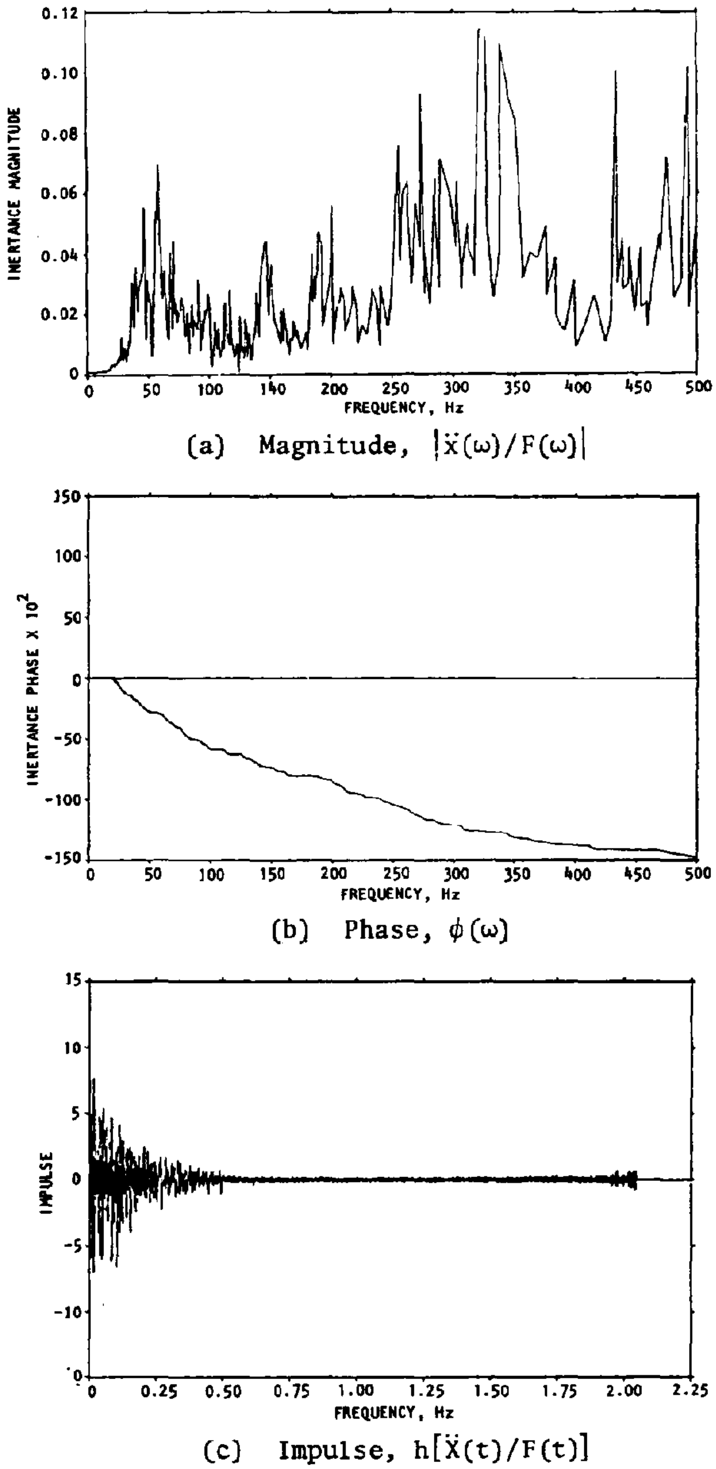


Figure 9-27. Platform PARPP-CR, Safeguard system: Typical Transfer Acceleration-Acceptance Function, from Pulser Attachment Point to Reference Acceleration Point #1 on Platform. (Safford-Walker, 1975)

Additionally, modeling time and computer costs can be substantial.

(3) An empirical method for predicting shock spectra response was developed for shock-isolated platforms for the Safeguard system (Safford-Walker, 1975). To predict shock response for a general class of rectangular shock-isolated platforms with a variety of equipment mounted thereon, only the following information is required:

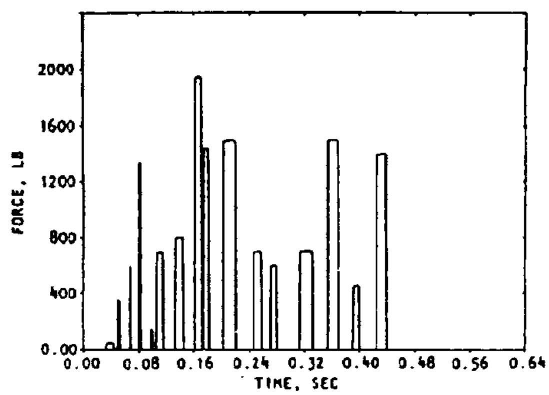
—Acceleration shock spectra of facility environment at location of shock-isolation system
—Length of platform
—Width of platform
—Average weight supported by each isolator
—Density: ratio of the total weight of platform and equipment to platform area

These empirical predictions have the following constraints:

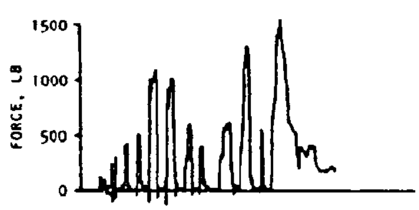
—Frequency range, 36 Hz to 460 Hz
—Vertical direction only
—Facility environment—oscillatory time history (with frequency components of 36 Hz to 460 Hz and an approximate time duration of 2 sec)

(4) A very simple computer routine can be made from the information presented in this section (see app. B).

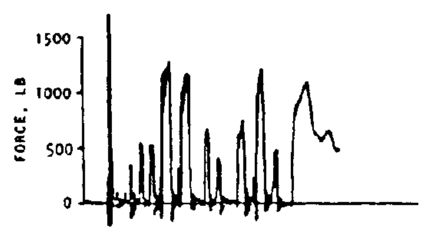
(5) A multivariate regression method was used to predict responses for shock-isolated platforms. Regression analysis may be viewed as the problem of determining a linear relationship between two or more variables. It may also be viewed as a linear curve-fitting procedure in which



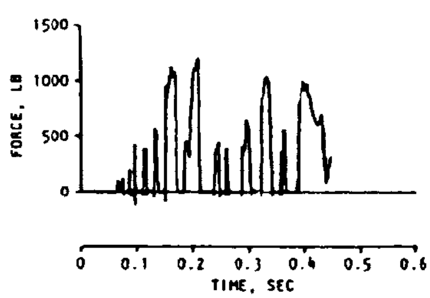
(a) Specified pulse train for each corner



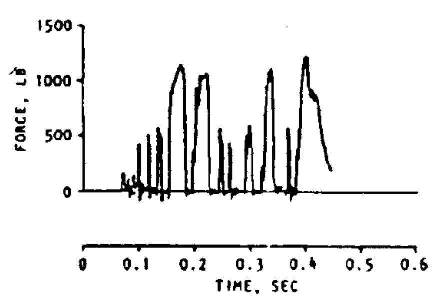
(b) Test pulses at SW corner



(c) Test pulses at SE corner

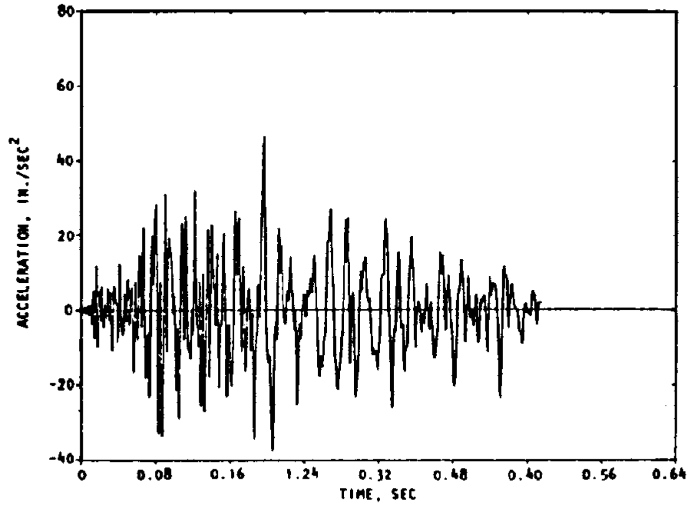


(d) Test pulses at NE corner

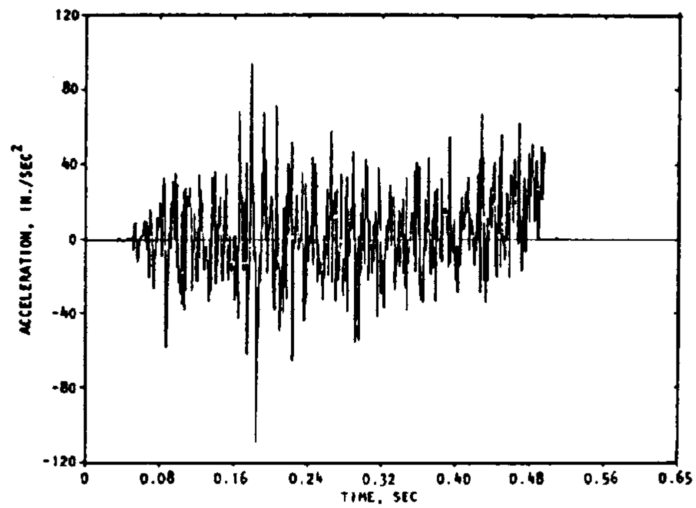


(e) Test pulses at NW corner

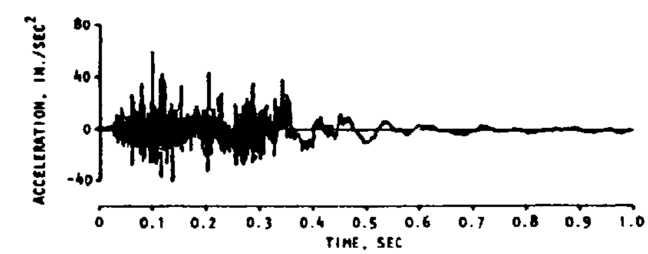
Figure 9-28. Platform PARPP-CR: Specified and Actual Test-Input Pulses (Safford-Walker, 1975)



(a) Predicted

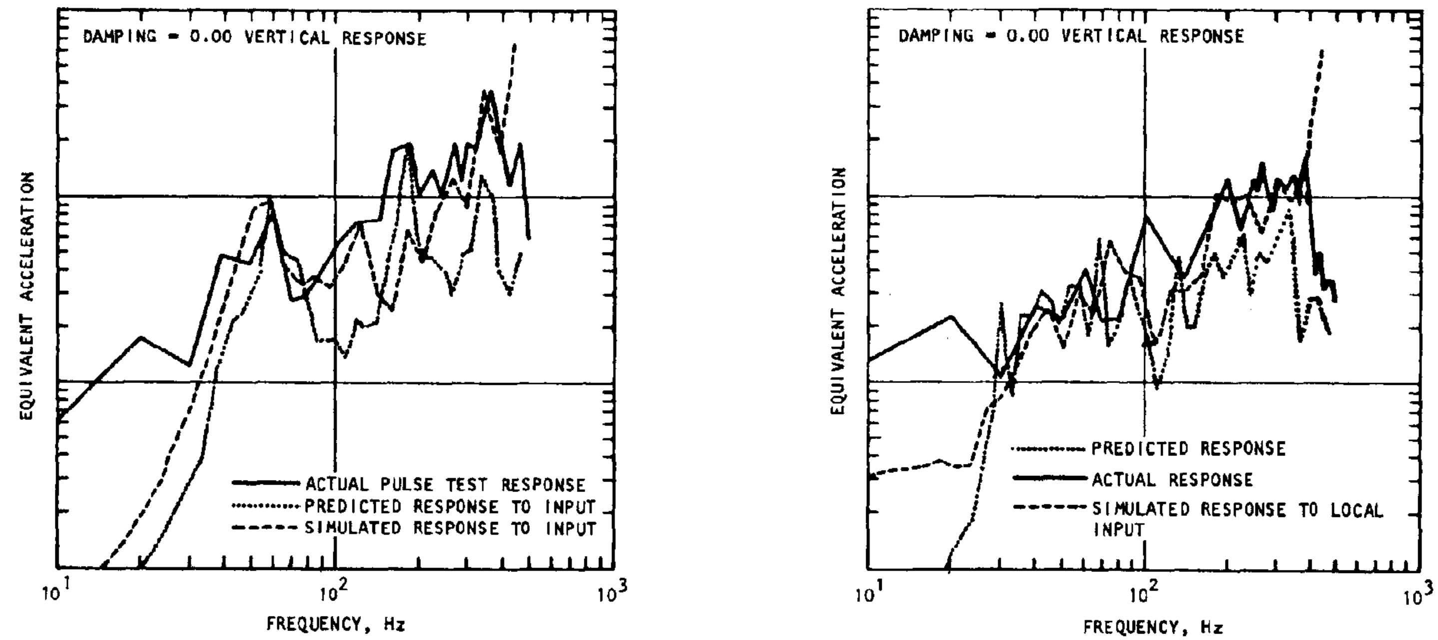


(b) Pulse simulated



(c) Pulse tested

Figure 9-29. Platform PARPP-CR, Safeguard System: Acceleration-Time Histories of Predicted, Pulse-Simulated, and Pulse-Tested Motions (Safford-Walker, 1975)



(a) Platform D

(b) Control room platform

Figure 9-30. Comparison of Shock Spectra for the Predicted, Pulse-Simulated, and Pulse-Tested Platforms, Safeguard System (Safford-Walker, 1975)

$$Y = \beta_0 + \beta_1 X_1 + \beta_2 X_2 + \cdots + \beta_n X_n \qquad (9\text{-}6)$$

where

Y = Dependent variable
β_n = Regression coefficients to be determined
X_n = Independent variables

(6) In the regression computation, values of β are assumed; and with the independent variables of X_n, Y is computed. This Y is compared to the known Y value for the platform, and the β values are adjusted through iterative procedures to within some error limit. The independent variables, X_n, are the particular characteristics of any platform.

(7) Examination of the multivariate regression procedure indicated that a polynomial set of independent variables was needed to generate predictions within acceptable error. The polynomial approach was generated by representing the shock-isolated platforms in the form of plates. Since the dynamic response of a plate (hence its shock spectrum) is directly related to its frequency spectrum, the following characteristic parameters were chosen as independent parameters for the regression analysis: ℓ, w, ρ, wt/iso. On the basis of several trial cases, the following nonlinear multivariate regression was evolved:

$$\ddot{x}(ss) = \beta_0 + \beta_1 \ell + \beta_2 \ell^2 + \beta_3 \ell^3 + \beta_4 \rho + \beta_5 \rho^2 + \beta_y \rho^3 + \beta_7(\ell/w) + \beta_8(wt/iso) \qquad (9\text{-}7)$$

where

$\ddot{x}(ss)$ = Acceleration shock spectra
β_n = Coefficients of regression for each frequency point
ρ = Weight/area, lb/ft^2 (calculated density)
ℓ = Length, ft
w = Width, ft
wt/iso = Total weight per isolator

(8) The particular data bases used for this analysis were the dependent variable values of shock spectra responses from 10 platforms. These platforms were measured in-place for transfer functions (figs. 5-13 to 5-22). Using a reference facility input motion (fig. 9-31), the acceleration time history and shock spectra response were calculated for each of the 10 platforms as covered in the procedures of paragraph 9-3. The independent variable platform characteristics—length, width, density, length/width ratio, and weight per isolator—are given in table 9-3. Note the wide range of these independent variables. To ensure optimum predictive results, the error limit for the response data for the 10 measured platforms was set at ± 1 standard deviation (± 1σ) about the mean. Also, because of the optimum distributive sampling of the measured platforms, the upper and lower bounds of most practical platforms will not exceed the upper and lower bounds of the 10 platforms under investigation (fig. 9-32). The regression coefficients determined for each shock spectrum frequency are listed in table 9-4.

(9) Figures 9-33 and 9-34 show two platform responses to the referent environment input as computed from transfer function data. Overlaid on each plot is the predicted shock spectrum for that platform, derived by multivariate regression analysis.

b. Predicted shock spectra.

(1) The predicted shock spectra show a good correlation with the actual shock spectra if all 10 platforms are viewed in a statistically functional relationship. Individually, 7 of the 10 platforms (in the data base) are almost perfectly predicted to within ± ¼ standard deviations.

(2) The shock spectrum responses of platforms can be predicted by using their specific characteristics (length, width, length/width, density, wt/iso) with the coefficients from table 9-4. These values are the platform responses to the reference input. Typical examples are responses predicted for unmeasured platforms by the regression method are shown in figures 9-35 and 9-36.

(3) To develop the platform shock spectrum for a particular facility location, a scaling technique is used, and it should comply reasonably with the constraints of wave shape similarity, frequency bandwidth, and time duration of the reference input. The procedure calls for dividing the shock spectrum values of the facility environments under consideration by the shock spectrum values of the reference environment. A relationship can be drawn as follows to form a pseudo-transfer function.

$$\ddot{y}_{new}^{(ss)} = \ddot{y}_{ref.}^{(ss)} \; \frac{\text{facility input ss}}{\text{ref. input ss}} \qquad (9\text{-}8)$$

where

facility input ss = Acceleration input shock spectrum specified for a particular facility under consideration
ref. input ss = Reference acceleration shock spectrum
$\ddot{y}_{new}^{(ss)}$ = Platform acceleration response shock spectrum to facility input environment
$\ddot{y}_{ref.}^{(ss)}$ = Platform acceleration response shock spectrum to reference input environment

(4) Figures 9-35 and 9-36 also show platform response to shock spectrum input environments of different facilities, determined from the above scaling technique.

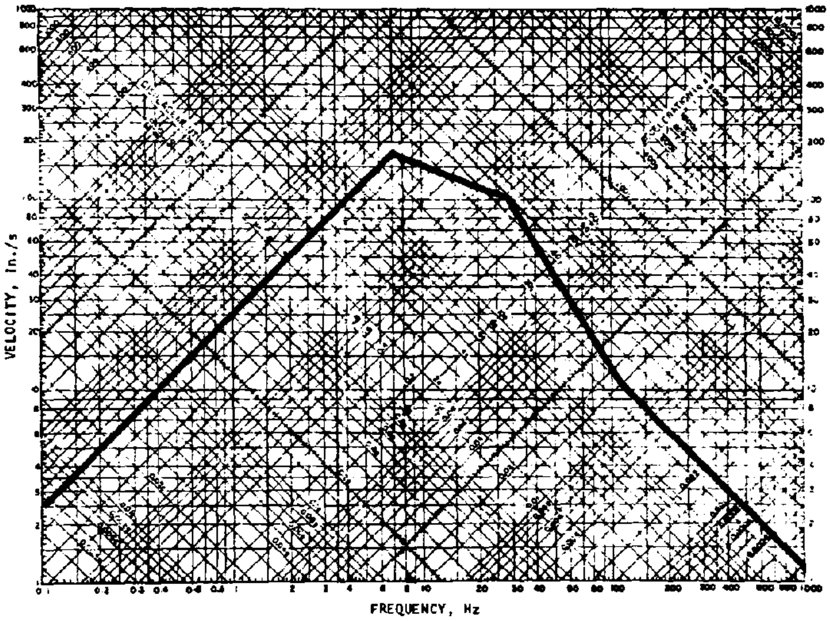
9-6. Time series data analysis.

a. Except for rather simple tests, the majority of validation work will require sophisticated time series data analysis. The reduction, processing, and presentation of test data is required to assess the hardness/vulnerability of shock isolation systems.

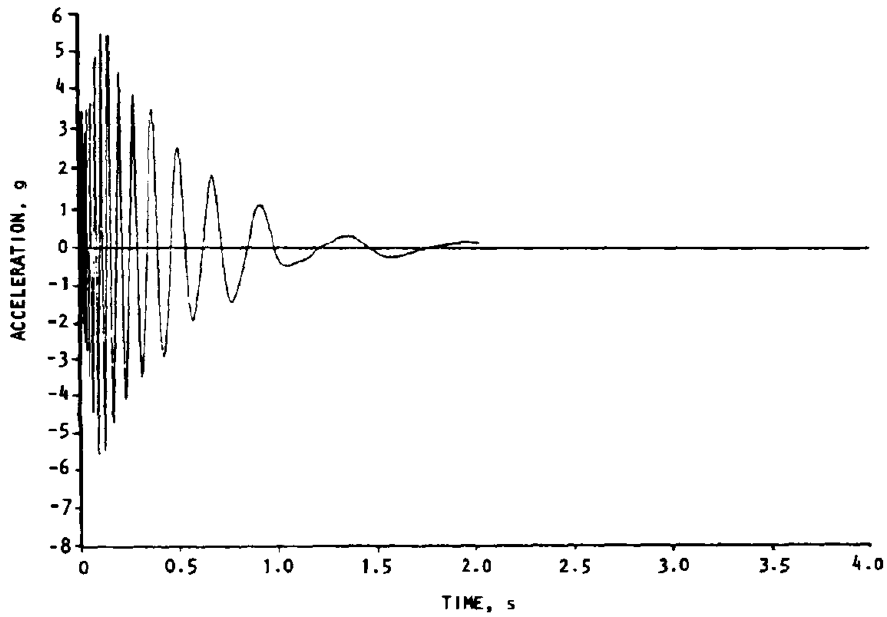
b. Basic methods of time series data analysis include the following:

(1) Amplitude Domain
Mean, variance, and rms
Probability density histogram

(2) Time Domain
Auto- and cross-covariance (correlation functions



(a) Reference shock spectrum



(b) Synthesized waveform acceleration time history to match reference shock spectrum (Yang, 1971)

Figure 9-31. Reference Acceleration Input Motion for Shock Isolation Systems (Safford-Tuttle, 1974)

TABLE 9-3. REGRESSION PARAMETERS FROM 10 TRANSFER-FUNCTION MEASURED PLATFORMS (DATA BASE)

Platform	Length, ℓ, ft	Width, w, ft	Total Weight, lb	Number of Isolators	ℓ/w	Density ρ, lb/ft^2	Weight per Isolator, lb/unit
PARPP-H	14	5	7,500	4	2.8	107	1,875
RLOB-124B	10	8	11,000	4	1.25	137	2,750
PARPP-D	12	4	13,100	4	3.0	273	3,275
PARPP-G	13	5.5	13,700	4	2.36	192	3,425
RLOB-124C	23	7	23,500	6	3.29	146	3,917
MSCB-125B	31	9	32,600	4	3.44	117	8,150
MSRPP-9-10	26	24	66,600	20	1.08	107	3,330
PARPP-A	52	14	109,300	17	3.71	150	6,429
PARPP-CR	64	49	198,000	18	1.31	63	11,000
MSRPP-2W	77	16	244,000	20	4.81	198	12,200

U.S. Army Corps of Engineers

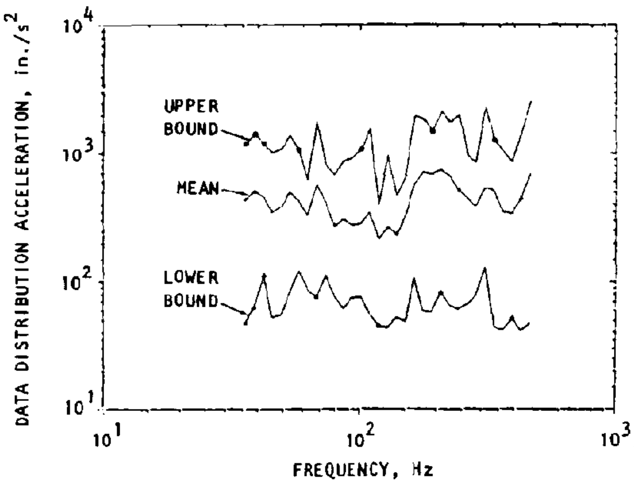


Figure 9-32. Distribution of Shock Spectra for 10 Platforms Measured in Place (Response Computations Determined from Reference Input.) (Safford-Walker, 1975)

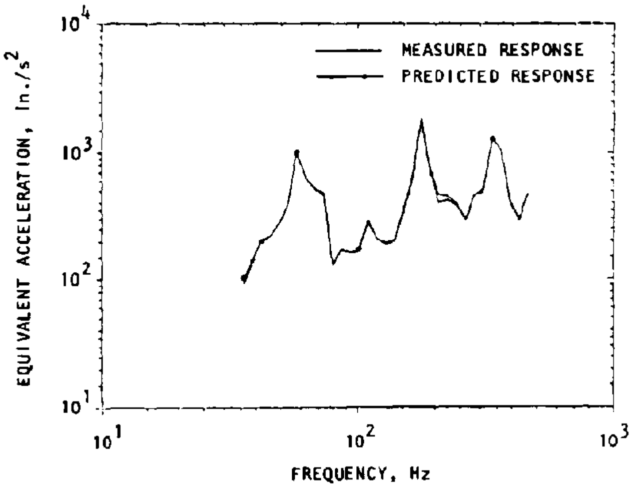


Figure 9-33. Platform PARPP-D, Safeguard System: Comparison of Shock Spectra Determined from Transfer Function Measurements and from Regression Prediction Method (Safford-Walker, 1975)

Convolution
Recursive digital filters
(3) Frequency Domain
Fourier transforms
Power (auto) and cross spectra
Frequency response functions
Coherence functions
(4) Miscellaneous
Digital filters

c. The above methods can be used for additional analyses, including the computation of system functions (impedance, inertance, transfer functions) and impulse functions, and the extraction of mode shapes. When used with input excitation, responses in the time and frequency domain may also be determined.

d. Computer hardware and software systems employed in the analysis of shock and vibration test data are fully discussed in *Shock and Vibration Computer Programs, Reviews and Summaries*, (Pilkey-Pilkey, 1975). The hardware and software systems are summarized in appendix B.

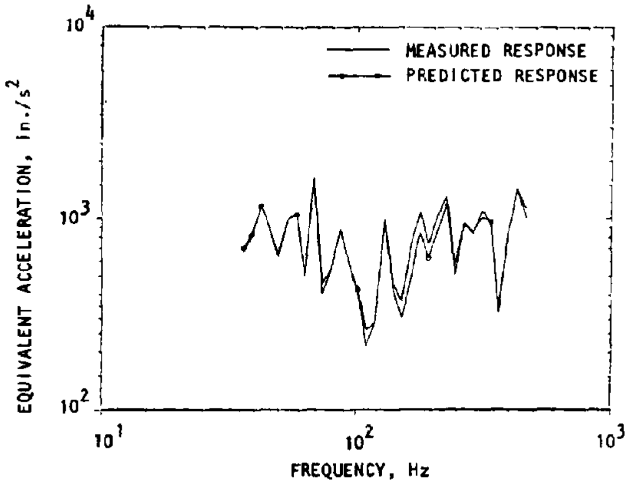


Figure 9-34. Platform PARPP-CR, Safeguard System: Comparison of Shock Spectra Determined from Transfer Function Measurements and from Regression Prediction Method (Safford-Walker, 1975)

TABLE 9-4. MULTIVARIATE REGRESSION COEFFICIENTS FOR PREDICTIVE ANALYSIS

β_0	β_1	β_2	β_3	β_4	β_5	β_6	β_7	β_8	Frequency, Hz
-0.3671+4	0.1816+3	-0.4627+1	0.3658-1	0.5376+2	-0.3186-0	0.6015-3	-0.3542+3	0.2769-1	0.3587+2
-0.3122+4	0.7055+2	-0.1866+1	0.1581-1	0.5248+2	-0.2986+0	0.5390-3	-0.1834+3	0.8508-1	0.3885+2
0.1324+4	0.3176+2	-0.7958+0	0.7189-2	-0.2768+2	0.1661+0	-0.3041-3	-0.5416+2	0.3755-1	0.4208+2
0.2770+3	0.2708+2	-0.7668+0	0.7888-2	0.3615+1	0.1999-1	-0.2525-4	-0.1168+3	0.2307-1	0.4557+2
-0.2231+4	0.1534+3	-0.3755+1	0.3092-1	0.3237+2	-0.1924+0	0.3828-3	-0.3151+3	-0.5636-1	0.4936+2
0.9328+3	0.4182+2	-0.7003+0	0.6436-2	-0.2307+2	0.1691+0	-0.3339-3	-0.1138+3	-0.3805-1	0.5346+2
0.1915+4	0.1321+1	-0.1115+0	0.2587-2	-0.2493+2	0.8769-1	-0.3439-4	0.1548+2	0.3273-2	0.5790+2
0.2106+3	0.2724+2	-0.4712+0	0.3393-2	-0.5182+1	0.3216-1	-0.2902-4	-0.7303+2	-0.9454-2	0.6272+2
0.9574+3	0.2593+3	-0.5747+1	0.4141-1	-0.3565+2	0.1926+0	-0.2729-3	-0.5078+3	-0.1049+0	0.6793+2
0.5243+3	0.1257+3	-0.2937+1	0.2139-1	-0.2981+2	0.2299+0	-0.4646-3	-0.2035+3	-0.3083-1	0.7357+2
0.1111+4	0.8103+1	0.7443-1	-0.8058-3	-0.2447+2	0.1699+0	-0.3424-3	-0.1392+2	-0.1855-1	0.7969+2
0.9228+3	0.1217+2	0.1296+0	-0.1420-2	-0.1416+2	0.8690-1	-0.1530-3	-0.1106+3	-0.2226-1	0.8631+2
-0.1158+3	-0.1533+2	0.5640+0	-0.2672-2	0.8084+1	-0.5302-1	0.1078-3	0.5595+1	-0.2430-1	0.9348+2
-0.1040+4	0.4503+2	-0.1289+1	0.1220-1	0.1445+2	-0.8032-1	0.1474-3	-0.5101+2	0.9394-2	0.1013+3
-0.4469+4	0.2538+3	-0.5807+1	0.4325-1	0.6331+2	-0.3757+0	0.7323-3	-0.4543+3	-0.1045+0	0.1097+3
-0.1325+4	0.1507+2	-0.2160+0	0.1230-2	0.2513+3	-0.1526+0	0.2944-3	-0.5427+2	0.2115-1	0.1188+3
0.2422+4	-0.2801+1	0.2792+0	-0.2370-2	-0.4278+2	0.2466+0	-0.4361-3	-0.2687+2	0.5897-2	0.1287+3
0.1932+4	0.1273+2	-0.1695+0	0.3801-3	-0.2216+2	0.1415+0	-0.2691-3	-0.3453+1	0.8266-2	0.1393+3
-0.1460+4	-0.1057+2	0.2085+0	-0.7054-4	0.3544+2	-0.2048+0	0.3754-3	-0.7009+2	0.1641-1	0.1509+3
0.2432+4	0.6671+2	-0.1154+1	0.5777-2	-0.5630+2	0.3595+0	-0.6852-3	0.9941+2	0.7622-1	0.1635+3
0.3143+4	0.8239+2	-0.9593+0	0.2640-2	-0.6250+2	0.3211+0	-0.4407-3	0.1289+3	-0.1567+0	0.1770+3
-0.5764+3	-0.2426+3	0.7021+1	-0.5304-1	0.7875+2	-0.5125+0	0.9827-3	0.1297+3	-0.1322+0	0.1918+3
-0.3351+4	-0.2592+3	0.6601+1	-0.4634-1	0.1428+3	-0.8571+0	0.1552-2	-0.2567+3	0.2296-1	0.2077+3
-0.7135+3	-0.1400+3	0.3136+1	-0.1885-1	0.7739+2	-0.4940+0	0.9336-3	-0.3281+3	0.2230-1	0.2249+3
-0.6098+4	-0.2125+3	0.4958+1	-0.3242-1	0.1876+3	-0.1111+1	0.2023-2	-0.5176+3	0.1051+0	0.2436+3
0.4951+4	0.3359+2	-0.9849+0	0.8110-2	-0.1019+3	0.6770+0	-0.1339-2	-0.1063+3	-0.4448-2	0.2639+3
0.2153+4	-0.1006+3	0.2945+1	-0.2299-1	-0.1525+2	0.8887-1	-0.1569-3	-0.2303+2	-0.2198-1	0.2858+3
0.1328+5	-0.4249+1	0.3202+0	-0.8384-2	-0.2834+3	0.1911+1	-0.3817-2	-0.7398+2	0.2743-1	0.3096+3
0.4553+4	-0.9148+2	0.2788+1	-0.2400-1	-0.7136+2	0.4701+0	-0.8750-3	-0.1155+3	0.1199-1	0.3353+3
0.2531+4	-0.1543+1	0.2966+0	-0.3936-2	-0.5116+2	0.3498+0	-0.6554-3	-0.7523+2	-0.1623-1	0.3632+3
0.5444+4	0.1029+2	0.6352-1	-0.2988-2	-0.1101+3	0.7199+0	-0.1397-2	-0.9570+2	-0.1380-1	0.3933+3
0.7325+4	-0.2826+2	0.1256+1	-0.1232-1	-0.1330+3	0.8424+0	-0.1617-2	-0.1001+3	-0.3762-1	0.4260+3
0.1263+5	-0.1144+3	0.4560+1	-0.4489-1	-0.2607+3	0.1829+1	-0.3742-2	-0.2691+2	-0.8684-1	0.4614+3

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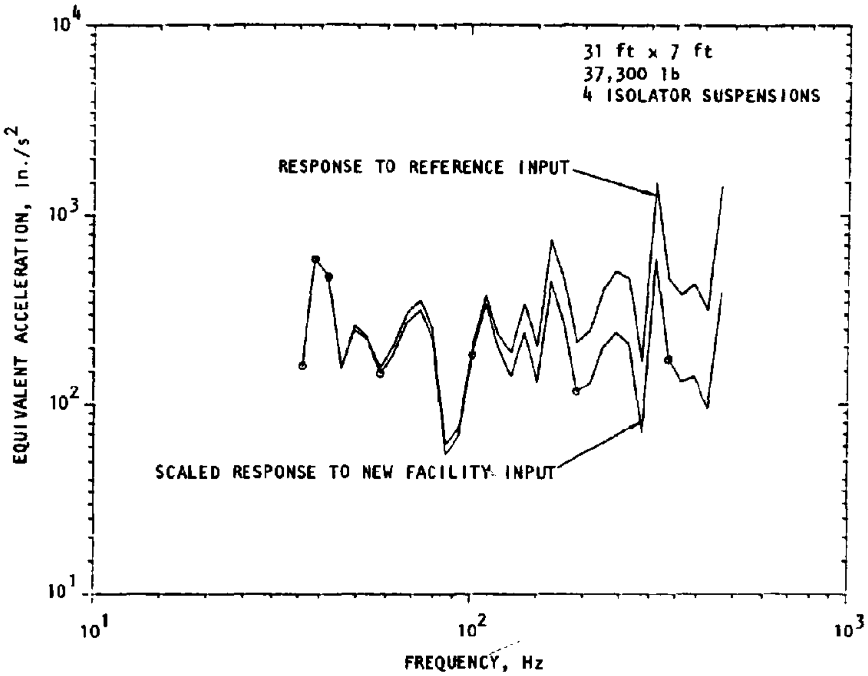

Figure 9-35. Platform MSCB-250A, Safeguard System: Vertical Shock-Response Spectra (Regression Prediction Method) (Safford-Walker, 1975)

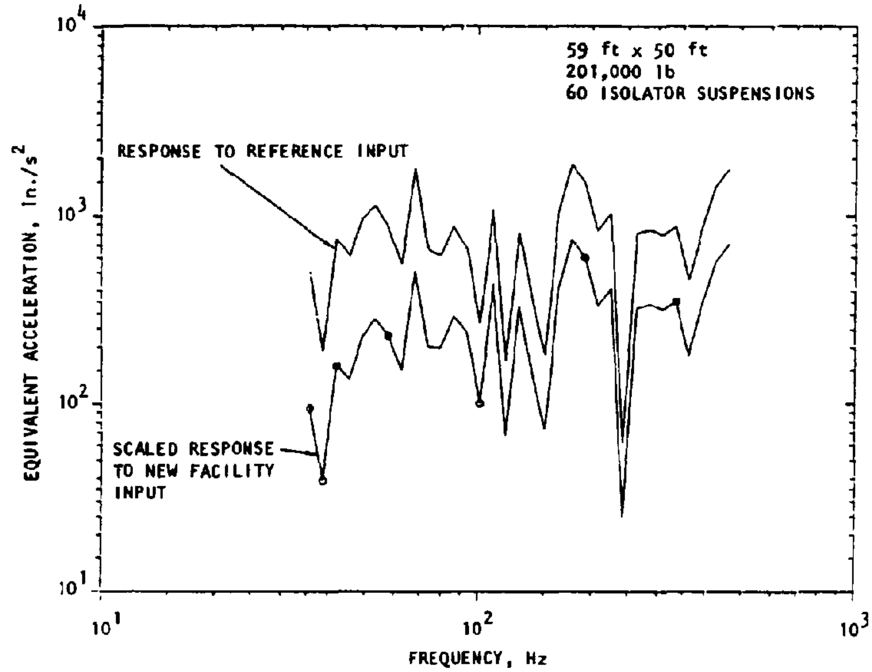

Figure 9-36. Platform MSRPP-1, Safeguard System: Vertical Shock-Response Spectra (Regression Prediction Method) (Safford-Walker, 1975)

APPENDIX A

TRANSIENT AND SHOCK TEST DATA FOR VARIOUS EQUIPMENTS

Available data from transient and shock tests performed upon various types of equipment are presented in this appendix. Summaries of tests conducted, test machines, types of equipment and test results are provided. Reference documents covering each equipment tested are given for each tabulation. The appendix is in five main divisions:

A-1. Equipment tests by U.S. Navy LWHI and MWHI shock machines
A-2. Transient shock tests of commercial communication equipment
A-3. Transient shock tests of commercial electric power and conditioning equipment
A-4. Shock spectrum analysis for command communication equipment
A-5. Shock testing for Emergency Operating Centers in Albany and Oklahoma City

Each division of this appendix deals with a unique environment and equipments. The test descriptions are general. Usage must rely on engineering judgment to guide the selection of those tests applicable to the specific shock isolation system and requirements of interest. References and credits listed at the end of each division of this appendix provide sources of additional information and identify sources on which figures and tables are based.

A-1. Equipment Tests by U.S. Navy LWHI and MWHI Shock Machines.

A-1-1. Shock response spectra of equipments tested upon U.S. Navy light-weight, high-impact (LWHI) and medium-weight, high-impact (MWHI) shock machines are provided in figures A-1-1 through A-1-13. MIL-S-901 is the specification covering the use of these machines. The response spectra are presented for a range of equipment weights, impact hammer drop heights, mounting plates, and specimen orientation on the machine. Shock tests performed with the lightweight machine normally utilize four types of mounts; these are designated the 4A, 4C, 6E, and 6D mountings. The first two are usually used when testing mechanical systems, and the latter two are used for electrical systems. Applications may be made by evaluation, interpretation, and interpolation of the data and the equipment.

A-1-2. Test results from equipment tested upon U.S. Navy LWHI and MWHI shock machines are presented in tables A-1-1 through A-1-43. An index to these tables is provided below:

EQUIPMENT-TO-TABLE CROSS REFERENCE

	Table
Electronic Equipment	
Antennas	A-1-1
Frequency Standard	A-1-2
Oscilloscopes	A-1-3
Oscilloscope Components	A-1-4
Time Comparator Systems	A-1-5
Tube Traveling Wave	A-1-6
Mechanical Equipment	
Compressor	A-1-7
Diesel-Generators (see under electrical eqipment)	
Fans	A-1-8
Fastener Anchor	A-1-9
Frame, Diesel Engine	A-1-10
Gages	A-1-11
Heat Exhanger	A-1-12
Motor and Pump	A-1-13
Pumps (see also motor and pump)	A-1-14
Shielding, Lead	A-1-15
Valves	A-1-16
Electrical Equipment	
Batteries	A-1-17
Battery Jars	A-1-18
Brush Holder	A-1-19
Bus Transfer Unit	A-1-20
Chokes (see oscilloscope components, Table A-1-4)	
Circuit Breakers	A-1-21
Contactors	A-1-22
Controller	A-1-23
Diesel Generators	A-1-24
Fittings, Encapsulated	A-1-25
Fuse Box	A-1-26
Gear Motor	A-1-27
Motors (see also motor and pump under mechanical equipment)	A-1-28
Generators	A-1-29
Motor Generators	A-1-30
Panels (Alarm, Control, and Indicator)	A-1-31
Plug and Receptacles	A-1-32
Rectifier	A-1-33
Relays	A-1-34
Resistors	A-1-35
Switchboards	A-1-36
Switches, Limit	A-1-37
Switches, Rotary	A-1-38
Switches, Rotary Snap	A-1-39
Swithces, Toggle	A-1-40
Switches, Miscellaneous	A-1-41
Temperature Elements	A-1-42
Voltage-Sensitive Elements	A-1-43

A-1-3. References and credits.

Dept. of Navy (NAVY). *Requirements for Shock Tests,* H.I. (High Impact); Shipboard Machinery, Equipment, and Systems, MIL-S-901 Washington, D.C.: Navy.

Eubanks, R. A. and Jaskie, B. R. "Shock Hardening of Equipment," *Shock and Vibration Bull.* No. 32, Pt. 3. Washington, D.C.: Naval Res. Lab, Dec 1963.

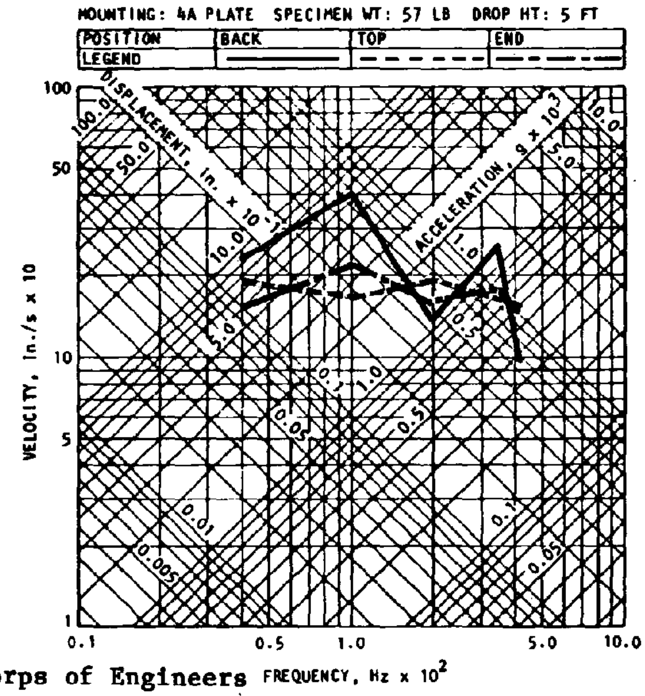

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FIGURE A-1-1. SHOCK SPECTRA FOR NAVY LIGHTWEIGHT, HIGH-IMPACT, SHOCK MACHINE (MOUNTING = 4A PLATE, SPECIMEN WEIGHT = 57 LB, DROP HEIGHT = 5 FT)

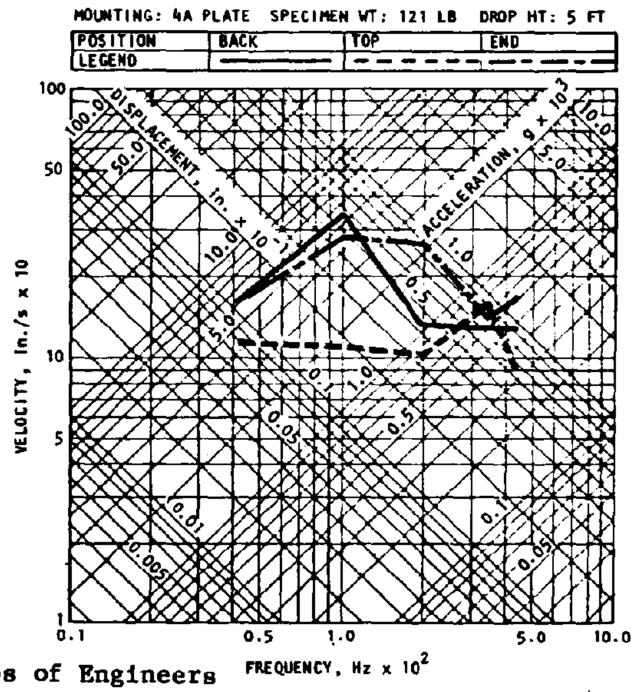

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FIGURE A-1-2. SHOCK SPECTRA FOR NAVY LIGHTWEIGHT, HIGH-IMPACT, SHOCK MACHINE (MOUNTING = 4A PLATE, SPECIMEN WEIGHT = 121 LB, DROP HEIGHT = 5 FT)

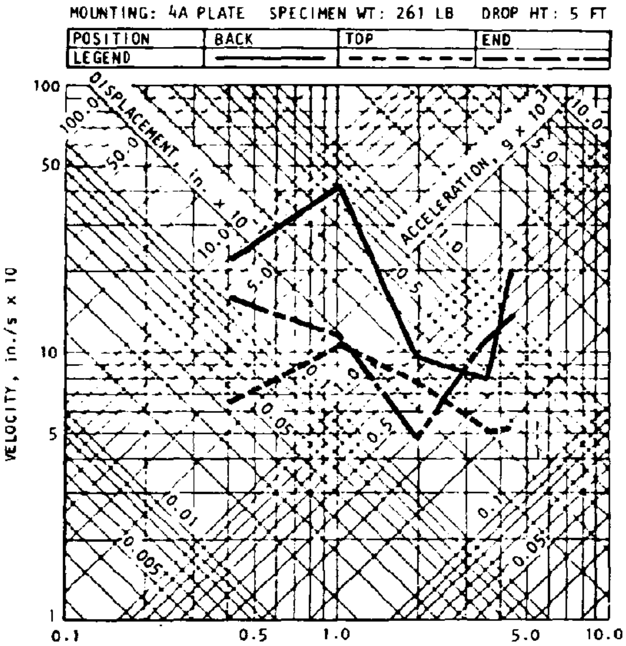

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FIGURE A-1-3. SHOCK SPECTRA FOR NAVY LIGHTWEIGHT, HIGH-IMPACT, SHOCK MACHINE (MOUNTING = 4A PLATE, SPECIMEN WEIGHT = 261 LB, DROP HEIGHT = 5 FT)

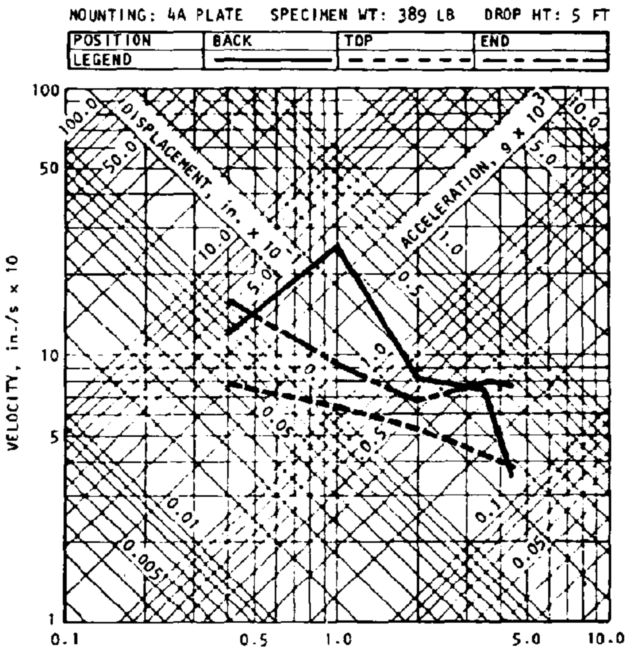

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FIGURE A-1-4. SHOCK SPECTRA FOR NAVY LIGHTWEIGHT, HIGH-IMPACT, SHOCK MACHINE (MOUNTING = 4A PLATE, SPECIMEN WEIGHT = 389 LB, DROP HEIGHT = 5 FT)

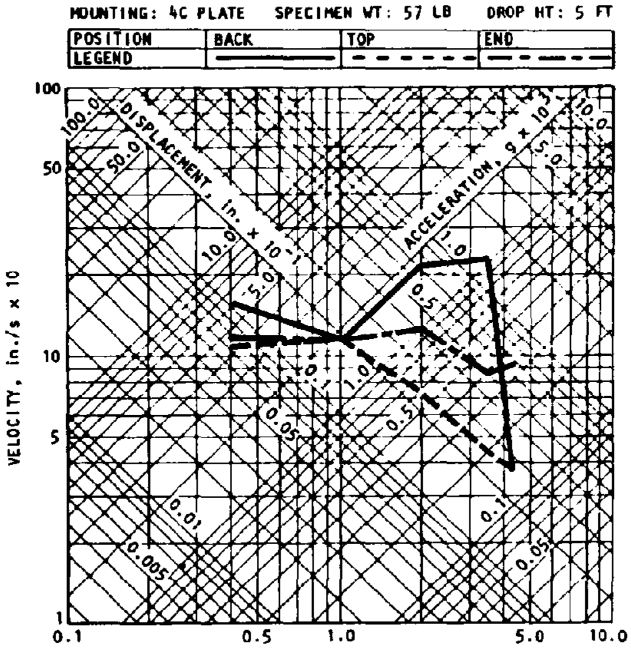

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FIGURE A-1-5. SHOCK SPECTRA FOR NAVY LIGHTWEIGHT, HIGH-IMPACT, SHOCK MACHINE (MOUNTING = 4C PLATE, SPECIMEN WEIGHT = 57 LB, DROP HEIGHT = 5 FT)

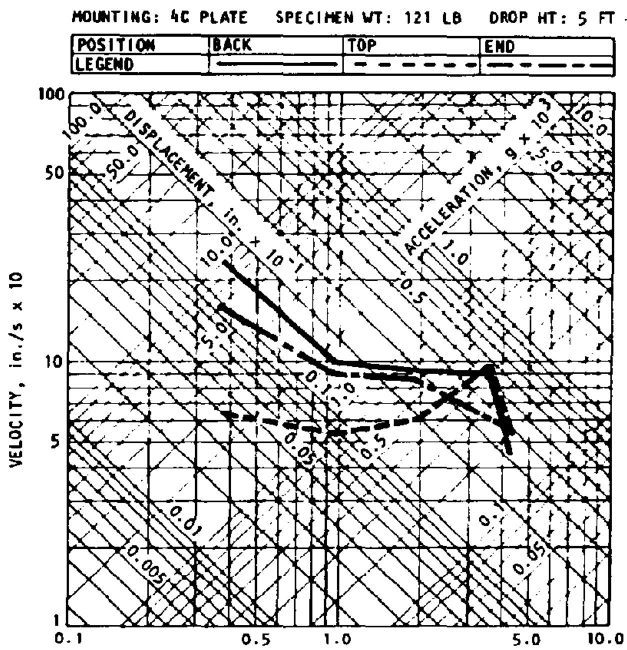

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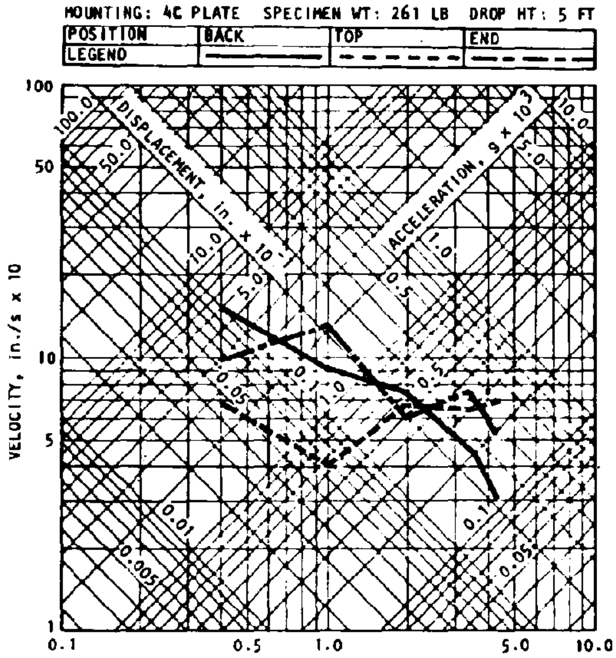
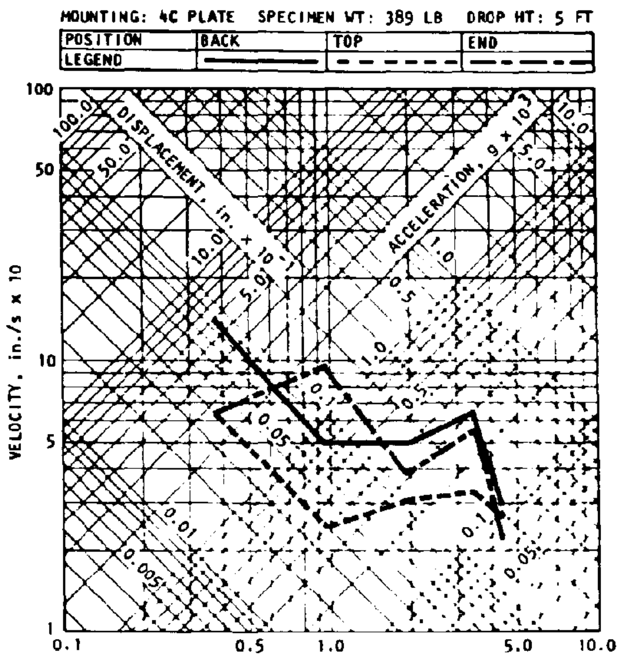
FIGURE A-1-6. SHOCK SPECTRA FOR NAVY LIGHTWEIGHT, HIGH-IMPACT, SHOCK MACHINE (MOUNTING = 4C PLATE, SPECIMEN WEIGHT = 121 LB, DROP HEIGHT = 5 FT)



U.S. Army Corps of Engineers FREQUENCY, Hz x 10^2

FIGURE A-1-7. SHOCK SPECTRA FOR NAVY LIGHTWEIGHT, HIGH-IMPACT, SHOCK MACHINE (MOUNTING = 4C PLATE, SPECIMEN WEIGHT = 261 LB, DROP HEIGHT = 5 FT)



U.S. Army Corps of Engineers FREQUENCY, Hz x 10^2

FIGURE A-1-8. SHOCK SPECTRA FOR NAVY LIGHTWEIGHT, HIGH-IMPACT, SHOCK MACHINE (MOUNTING = 4C PLATE, SPECIMEN WEIGHT = 389 LB, DROP HEIGHT = 5 FT)

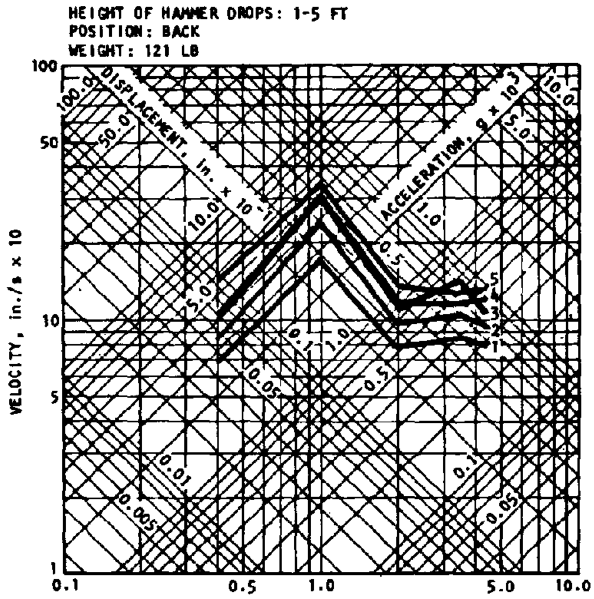


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FIGURE A-1-9. SHOCK SPECTRA FOR NAVY LIGHTWEIGHT, HIGH-IMPACT, SHOCK MACHINE (HEIGHT OF HAMMER DROPS = 1-5 FT, POSITION = BACK, WEIGHT = 121 LB)

GROUP	I	II	III
HEIGHT OF HAMMER DROP	1.0 FT	2.0 FT	2.0 FT
TABLE TRAVEL	3.0 IN.	3.0 IN.	1.5 IN.
LEGEND	———	— — —	—-—-—

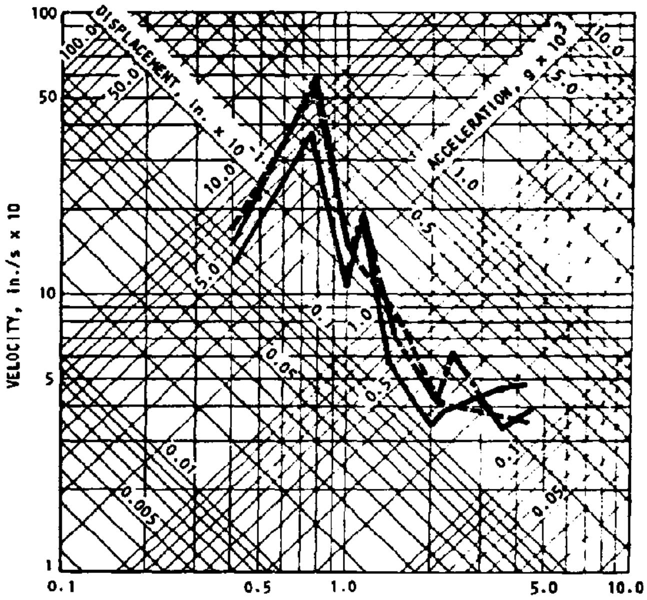

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FIGURE A-1-10. CLASS A TEST SPECTRA FOR NAVY MEDIUM-WEIGHT, HIGH-IMPACT, SHOCK MACHINE (EQUIPMENT WEIGHT = 1115 LB, TOTAL WEIGHT ON ANVIL TABLE = 1858 LB)

GROUP	I	II	III
HEIGHT OF HAMMER DROP	1.5 FT	2.5 FT	2.5 FT
TABLE TRAVEL	3.0 IN.	3.0 IN.	1.5 IN.
LEGEND			

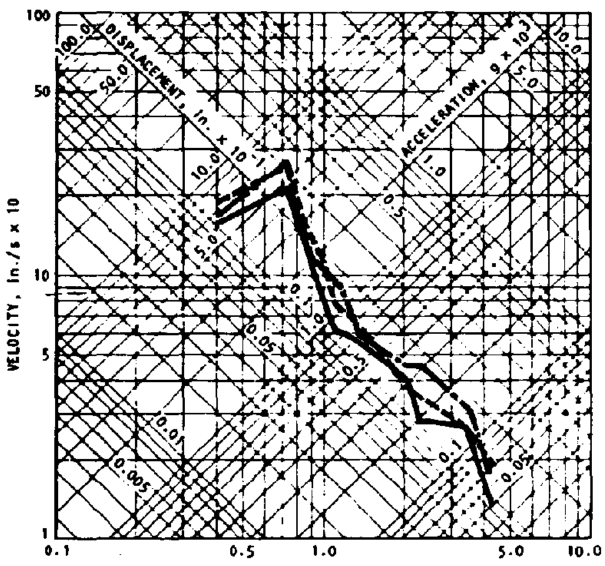

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FIGURE A-1-11. CLASS A TEST SPECTRA FOR NAVY MEDIUM-WEIGHT, HIGH-IMPACT, SHOCK MACHINE (EQUIPMENT WEIGHT = 2051 LB, TOTAL WEIGHT ON ANVIL TABLE = 3026 LB)

GROUP	I	II	III
HEIGHT OF HAMMER DROP	2.0 FT	3.5 FT	3.5 FT
TABLE TRAVEL	3.0 IN.	3.0 IN.	1.5 IN.
LEGEND			

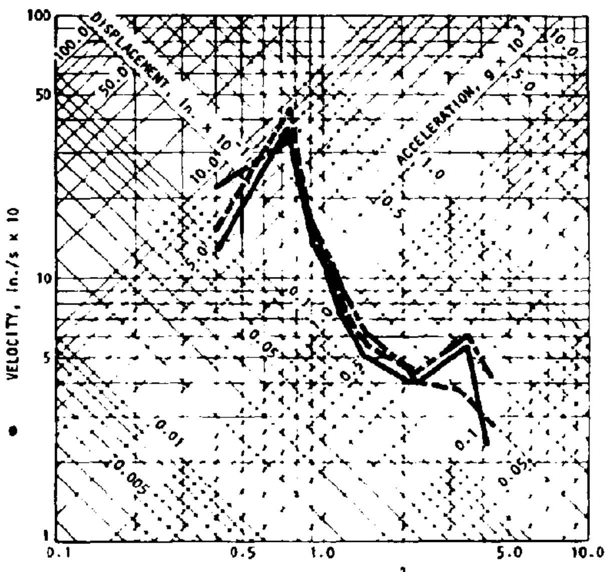

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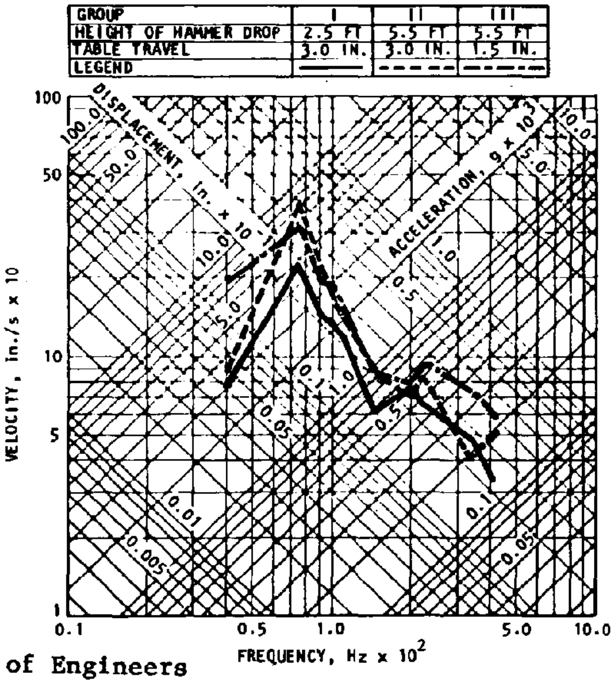
FIGURE A-1-12. CLASS A TEST SPECTRA FOR NAVY MEDIUM-WEIGHT, HIGH-IMPACT, SHOCK MACHINE (EQUIPMENT WEIGHT = 3386 LB, TOTAL WEIGHT ON ANVIL TABLE = 4424 LB)



U.S. Army Corps of Engineers

FIGURE A-1-13. CLASS A TEST SPECTRA FOR NAVY MEDIUM-WEIGHT, HIGH-IMPACT, SHOCK MACHINE (EQUIPMENT WEIGHT = 4423 LB, TOTAL WEIGHT ON ANVIL TABLE = 5391 LB)

TABLE A-1-1. ANTENNAS

General Description: RCM antennas, two Type AS-391, two Types AT-693, and 1 Type 994 were reported on.

Type	Approximate Weight (lb)	Test Environment	Test Machine	Number Tested	Test Specification	Report Designation	Result	Type of Failure
Antennas	30	1-, 3-, and 5-ft drops	Navy LWHI shock machine	5	MIL-S-901B 4C mounting	Mare Island NavShipYd 4054-61	Passed	—

TABLE A-1-2. FREQUENCY STANDARD

General Description: Model 103AR frequency standard.

Type	Approximate Weight (lb)	Test Environment	Test Machine	Number Tested	Test Specification	Report Designation	Result	Type of Failure
Model 103AR	65.5	1-, 3-, and 5-ft side blow; 1- and 3-ft top blow	Navy LWHI shock machine	1	MIL-E-16400	Mare Island NavShipYd 5569-50	Failed (3 ft top)	Crystal assembly was damaged

TABLE A-1-3. OSCILLOSCOPE

General Description: 1. Type AN/USN-105 oscilloscope with plug-in units.
2. Model 160AN oscilloscope with dual trace amplifier.
3. Type ANT/USN-105 oscilloscope.

Type	Approximate Weight (lb)	Test Environment	Test Machine	Number Tested	Test Specification	Report Designation	Result	Type of Failure
1	98	1-, 2-, and 3-ft and 2-, 3-, and 4-ft blows from back and side, and from top	Navy LWHI shock machine	1	MIL-S-901B 4C mounting	Mare Island NavShipYd 1806-60	Passed	—
2	72	1-, 2-, and 3-ft and 2-, 3-, and 4-ft blows from back and and side and from top	Navy LWHI shock machine	1	MIL-T-945 4C mounting	Mare Island NavShipYd 2365-59	Passed	—
3	90	1-, 2-, and 3-ft and 2-, 3-, and 4-, 1-, 2-, and 3-ft blows	Navy LWHI shock machine	1	MIL-S-901B 4C mounting	Mare Island NavShipYd 8103-59	Passed	—

TABLE A-1-4. OSCILLOSCOPE COMPONENTS

General Description: The components consisted of one transformer and four chokes.

Type	Approximate Weight (lb)	Test Environment	Test Machine	Number Tested	Test Specification	Report Designation	Result	Type of Failure
Components	30	1-, 3-, and 5-ft drops	Navy LWHI shock machine	5	MIL-S-901B 4A mounting	Mare Island NavShipYd 7810-59	Passed	—

TABLE A-1-5. TIME COMPARATOR SYSTEM

General Description: 1. Time comparator system HP-K-17-999B with frequency divider model 116AR.
2. Time comparator system HP-K17-999 A/B component with H13-120AR scope and 725 AR power supply.
3. Frequency divider and clock model 113X.

Type	Approximate Weight (lb)	Test Environment	Test Machine	Number Tested	Test Specification	Report Designation	Result	Type of Failure
1	144	1-, 2-, and 3-ft drops	Navy LWHI shock machine	2	MIL-S-901B 4C mounting	Mare Island NavShipYd 6987-61	Passed	—
2	180	1-, 2-, and 3-ft drops	Navy LWHI shock machine	1	MIL-S-901B 4C mounting	Mare Island NavShipYd 4296-61	Failed (1 ft back)	Cathode ray tube broke away from base
3	40	1-, 2-, and 3-ft drops	Navy LWHI shock machine	1	MIL-S-901B 4C mounting	Mare Island NavShipYd 1027-59	Passed	—

TABLE A-1-6. TRAVELING WAVE TUBE

General Description: Traveling wave tube No. X29.

Type	Approximate Weight (lb)	Test Environment	Test Machine	Number Tested	Test Specification	Report Designation	Result	Type of Failure
Tube	35	1-, 3-, and 5-ft drops	Navy LWHI shock machine	1	MIL-S-901B 4C mounting	Mare Island NavShipYd 4055-61	Failed (3 ft back)	Broken cathode

TABLE A-1-7. COMPRESSORS

General Description: The compressor was a two-stage, horizontally opposed, type AS150-300CFM air compressor. The compressor was driven by a gasoline engine, incorporated into the same unit as was the compressor. The compressor was tested with power on.

Type	Approximate Weight (lb)	Test Environment	Test Machine	Number Tested	Test Specification	Report Designation	Result	Type of Failure
150-300 CFM	210	1-, 3-, and 5-ft drop	Navy MWHI shock machine	1	MIL-S-901	ASTIA-AD 201782	Passed	–

TABLE A-1-8. FANS

General Description: Two reports were reviewed. Both fans were of 8-in. diameter. One used a 115-volt dc motor. The second used a 115-volt ac motor. Both fans are used for refrigeration and both were tested with power on.

Type	Approximate Weight (lb)	Test Environment	Test Machine	Number Tested	Test Specification	Report Designation	Result	Type of Failure
115-volt ac	60	1-, 3-, and 5-ft drops	Navy LWHI shock machine	1	JAN-F-151H	ASTIA-AD 43519	Failed	Base plate buckled
115-volt dc	60[a]	1-, 3-, and 5-ft drops	Navy LWHI shock machine	1	JAN-F-151H	ASTIA-AD 42119	Failed	Fan blade locked against guard

[a]Estimated

TABLE A-1-9. ANCHOR FASTENERS

General Description: The anchor fasteners consisted of an assembly of cable hangers and lengths of cable attached to a steel plate by means of a perforated surface plate and adhesive.

Type	Approximate Weight (lb)	Test Environment	Test Machine	Number Tested	Test Specification	Report Designation	Result	Type of Failure
Anchor	10	1-, 3-, and 5-ft drops	Navy LWHI shock machine	1	MIL-S-901B mounting not mentioned	Mare Island Report 1202-60	Failed on 5-ft drop back	Separation of 3/4-in. cable at the point of the "V."

TABLE A-1-10. DIESEL ENGINE FRAME

General Description: The equipment consisted of a diesel engine frame with two cylindrical castings and bearing cap assemblies. A bar of rolled steel was inserted into the crank shaft position to simulate the mass of the crankshaft. This was a single throw, type FV-9A diesel engine frame made of nodular iron.

Type	Approximate Weight (lb)	Test Environment	Test Machine	Number Tested	Test Specification	Report Designation	Result	Type of Failure
FV-9A	4565	2.0-, 3.5-, and 3.5-ft blows	MWHI shock machine	1	MIL-S-901	EES 5C101782	Passed	–

TABLE A-1-11. GAGES

General Description: The filter gage model MQD, which gives a signal when air filters should be serviced, was being considered for use by the Navy. The Navy also considered using a Magnehelic gage to indicate when a filter required cleaning. The pointer gage has slack diaphragms for sensing elements; the pointers are attached to the diaphragm by linkages. Two brass based pressure controls for air conditioning control equipment were also in the available reports.

Type	Approximate Weight (lb)	Test Environment	Test Machine	Number Tested	Test Specification	Report Designation	Result	Type of Failure
Filter	1.9	1- and 3-ft drop	Navy LWHI shock machine	1	MIL-S-901B 6D mounting	ASTIA-AD 160612	Failed 5th blow (back)	a
Magnehelic	1.1	1- and 3-ft drop	Navy LWHI shock machine	1	MIL-S-901B 6D mounting	ASTIA-AD 160458	Failed 6th blow (back)	b
Pointer	6.8	1-, 3-, and 5-ft drop	Navy LWHI shock machine	1	MIL-S-9018 6D mounting	Material lab NY NavShipYd 5956-3	Failed 1st blow Fixed, then passed	c
Control Sample A	4.6	1- and 3-ft drop	Navy LWHI shock machine	2	N.D. 66S3 4A mounting	Material lab NY NavShipYd 4429-C-22B	Failed	Cover clip loosened, lost adusting knob
Sample B	4.4	1-, 3-, and 5-ft drop						

[a]Entire front housing of the gage flew off and fell to the floor.
[b]Indicator needle broke off, helical bar supporting needle jumped its bearings, and the magnet broke.
[c]On the first blow, the indicator failed to indicate properly. On the fifth blow, the flame arrestor pointer failed to indicate properly. After calibration and repair of a cracked air hose, the equipment passed.

TABLE A-1-12. HEAT EXCHANGER

General Description: A laboratory-developed, water-cooled, heat exchanger was made to alleviate the problem of maintaining satisfactory ambient temperatures for electronic equipment aboard Naval vessels. The heat exchanger was filled with water during the shock test.

Type	Approximate Weight (lb)	Test Environment	Test Machine	Number Tested	Test Specification	Report Designation	Result	Type of Failure
Heat ex-changer	75	1-, 3-, and 5-ft drops	Navy LWHI shock machine	1	MIL-S-901B 4A mounting	Material lab NY NavShipYd 5441-3	Passed	–

TABLE A-1-13. MOTORS AND PUMPS

General Description: Eleven reports were reviewed on motors and pumps. It is most convenient for the equipment to be broken down into three classifications by weight. The equipment was tested running and at a standstill during alternate blows. The first class was the 0 to 300-lb class:

I
1. 1-hp, 30-gpm close-coupled centrifugal M and P
2. 3-hp, 250-gpm close-coupled centrifugal M and P
3. 5-hp, 50-gpm close-coupled centrifugal M and P
4. 2-hp, 2-gpm close-coupled centrifugal M and P

Next was the 300- to 600-lb class:

II
1. 3-hp, 100-gpm close-coupled centrifugal M and P
2. 15-hp, 185-gpm close-coupled centrifugal M and P
3. 5-hp, 400-gpm close-coupled centrifugal M and P
4. 10-hp, 125-gpm close-coupled centrifugal M and P
5. 7.5-hp, 600-gpm close-coupled centrifugal M and P

The final group consisted of those reports reviewed in the above 600-lb class:

III
1. 15-hp, 100-gpm close-coupled centrifugal M and P
2. 25-hp, 250-gpm close-coupled centrifugal M and P

Type	Approximate Weight (lb)	Test Environment	Test Machine	Number Tested	Test Specification	Report Designation	Result	Type of Failure
I	0-300	1-, 3-, and 5-ft drops	Navy MWHI shock machine	4	N.D. 6653	ASTIA-AD 201017 201031 201015 201016	4 passed	a
II	300-600	1-, 3-, and 5-ft drops	Navy MWHI shock machine	5	N.D. 6653	200977 201019 ATI205436 AD 201020 201018	4 passed 1 failed	b
III	600---	0.75-, 1.75-, and 1.75-ft blows	Navy MWHI shock machine	2	N.D. 6653	AD 201076 201077	Passed	

[a]On the 3-ft back blow, the 5-hp, 50-gpm motor and pump sustained permanent deformation of the shaft to bind during rotation.
[b]On the 1-ft front blow, the motor of the 5-hp, 400-gpm motor and pump sustained electrical damage. All of the above pumps had to be modified for whipping action of the overhanging mass at the pump housing.

TABLE A-1-14. PUMPS

General Description: 1. A positive displacement, constant delivery, vane type, hydraulic pump with a maximum fluid delivery rate of 8.4 gpm at 1200 rpm.
2. A 100-hp motor and suitable reduction gear as the prime mover. It too was a hydraulic pump, and was used as a power pump with a delivery rate of 192 gpm at 1100 psi.
The equipment was running at a standstill during alternate blows.

Type	Approximate Weight (lb)	Test Environment	Test Machine	Number Tested	Test Specification	Report Designation	Result	Type of Failure
Vane	29	1- and 3-ft drops	Navy MWHI shock machine	1	MIL-P-17869 4C mounting	ASTIA-AD 77814	Failed	Mounting base fractured on 3-ft vertical blow
Power	2575	1.25-, 2.25-, and 2.25-ft drops	Navy MWHI shock machine	1	MIL-S-901B	ASTIA-AD 159766	Passed	–

TABLE A-1-15. LEAD SHIELDING

General Description: Two types or methods of shielding surfaces with lead were reviewed; the first was by spraying the lead, and the second method was by burning in the lead.

Type	Approximate Weight (lb)	Test Environment	Test Machine	Number Tested	Test Specification	Report Designation	Result	Type of Failure
Sprayed	13.85	1-, 3-, and 5-ft drops	Navy LWHI shock machine	1	MIL-S-901B 4A mounting	Mare Island NavShipYd 1635-59	Failed	Hairline crack of bond
Burned in	14.0	1-, 3-, and 5-ft drops	–	1	MIL-S-901B 4A mounting	–	Passed	

TABLE A-1-16. VALVES

General Description:
1. Two ball valves.
2. A rotor valve.
3. Two 1/2-in., packless valves in closed position.
4. A pressure control, diaphragm operated external air pilot actuated type I valve (corresponds to a Navy type I, Series 150, Class B valve).

Type	Approximate Weight (lb)	Test Environment	Test Machine	Number Tested	Test Specification	Report Designation	Result	Type of Failure
Ball valves 2-1/2 in.	80	1-, 3-, and 5-ft drops	Navy LWHI shock machine	2	MIL-S-901B 4A mounting	Mare Island NavShipYd 4035-61	Passed	–
6 in.	30							
Rotor valve	75	1-, 3-, and 5-ft drops	Navy LWHI shock machine	1	MIL-S-901B 4A mounting	Mare Island NavShipYd 3188-61	Passed	–
Packless No. 1 and No. 2	3.5	1-, 3-, and 5-ft drops	Navy LWHI shock machine	2	Norman Browning Co. (pipe and metal blocks mounting)	Mare Island NavShipYd 1478-60	Passed	–
Pilot valves alum steel	10[a]	1-, 3-, and 5-ft drops	Navy LWHI shock machine	2	MIL-S-901B (mounted in piping)	ASTIA-AD 106139	Passed	–

[a]Estimated

TABLE A-1-17. BATTERIES

General Description:
1. Three samples of the Mk 47 Mod 0, silver oxide — zinc alkaline secondary type.
2. Two samples of the Mk 39 Mod 0 silver oxide — zinc alkaline secondary type in the dry charged condition.
3. Two preproduction samples of Battery Mk 39 Mod 0.
4. Two preproduction samples of Battery Mk 42 Mod 0 of the silver oxide — zinc alkaline secondary type.
5. Two production samples of Torpedo Propulsion Battery Mk 41 Mod 1.

Type	Approximate Weight (lb)	Test Environment	Test Machine	Number Tested	Test Specification	Report Designation	Result	Type of Failure
Mk 47	210	155 g	21-in. air gun	3	MIL-B-16955	ASTIA-AD 101822	Failed	Top cover seal failed
Mk 39	29.4	50 g for 0.05-sec duration	21-in. air gun	2	MIL-B-16955	ASTIA-AD 99715	Passed	–
Mk 39 pre-production	30.75	50 g for 0.05-sec duration	21-in. air gun	2	MIL-B-16955	ASTIA-AD 96344	Passed	–
Mk 42	64.0	100 g, 200 g for 0.04-sec duration	Torpedo cylinder	2	MIL-B-173484	ASTIA-AD 108996	Passed	–
Mk 41 Mod 1	18.5	50 g for 0.05 sec. Duration not mentioned	Bush-clevite torpedo battery mount	2	MIL-B-17348A	ASTIA-AD 151664	Passed	–

TABLE A-1-18. JARS (BATTERY)

General Description: 1. Two laminated hard rubber jars with type DRX-47 elements.
2. Four laminated hard rubber jars with concave exterior bottom and with type VSK-45 Exide type elements.
3. Two permali (laminated plywood) jars with Exide type 5350-I elements.

Type	Approximate Weight (lb)	Test Environment	Test Machine	Number Tested	Test Specification	Report Designation	Result	Type of Failure
Laminated hard rubber	92	1-, 2-, 3-, 4-, and 5-ft drops	Navy MWHI shock machine	2	EES-5B17 X0609	EES-5B066832	Passed	–
Concave bottom	90	1-, 2-, 3-, 4-, and 5-ft drops	Navy MWHI shock machine	4	EES-5B17 X0609	EES-5C066832	Passed	–
Permali jars	79	1-, 2-, 3-, 4-, and 5-ft drops	Navy MWHI shock machine	2	EES-5B17 X0609	EES-5D066832	Failed	Sustained slight cracks

TABLE A-1-19. BRUSH HOLDERS

General Description: The dc brush holder is of the reaction die cast holder type. The material is aluminum bronze.

Type	Approximate Weight (lb)	Test Environment	Test Machine	Number Tested	Test Specification	Report Designation	Result	Type of Failure
Dc brush holder	2	5 to 2000 ft-lb shocks	Navy LWHI shock machine	1	17G7	ASTIA-AD 59694	Passed	–

TABLE A-1-20. BUS TRANSFER UNIT

General Description: 1. Type ABT-A2 unit designed for automatic or manual operation (A 100/150-amp, 440-volt ac, 3-phase, 60 cycle).
2. Type ABT-A3 unit rated at 150-amp, 440-volt, 3-phase, 60-cycle, designed for automatic or manual operation.

Type	Approximate Weight (lb)	Test Environment	Test Machine	Number Tested	Test Specification	Report Designation	Result	Type of Failure
ABT-A2 unit	100[a]	1-, 3-, and 5-ft drops	Navy LWHI shock machine	1	MIL-S-901B 4A mounting	ASTIA-AD-205142	Passed	–
ABT-A3	150[a]	1-, 3-, and 5-ft drops	Navy LWHI shock machine	1	MIL-S-901B 4A mounting	ASTIA-AD-161434	Passed	–

[a]Estimated.

TABLE A-1-21. CIRCUIT BREAKERS

General Description: 1. Navy Type AQB, 3-pole, 400-amp frame, 500-volt ac, 250-volt dc units.
2. A type AQB-A250 breaker with a 160-amp trip unit.
3. A 1600-amp frame size with a 2-pole, 250-volt dc breaker.
4. A 2000-amp frame size type ACB.
5. Type ALB-1 circuit breakers-single pole, 50-amp frame size, 125-volt ac-dc units.

Type	Approximate Weight (lb)	Test Environment	Test Machine	Number Tested	Test Specification	Report Designation	Result	Type of Failure
400 amp	49.5	1-, 3-, and 5-ft drops	Navy LWHI shock machine	1	MIL-S-901 Special mounting	ASTIA-AD-207199	Failed 5 ft, rear	The breaker tripped
160 amp	25[a]	1-, 3-, and 5-ft drops	Navy LWHI shock machine	1	MIL-S-901	ASTIA-AD-141256	Failed 5 ft, vert.	Handle moved from closed position to open position
1600 amp	323	1-, 3-, and 5-ft drops	Navy LWHI shock machine	1	17B1(INT)	ASTIA-AD-435114	Passed Fig. 8 mounting	–
2000 amp	400[a]	0.75-, 1.75-, and 1.75-ft drops	Navy LWHI shock machine	1	MIL-S-901	ASTIA-AD-206127	Failed 5 ft, rear	The breaker tripped
50 amp	0.5	1-, 3-, and 5-ft drops	Navy LWHI shock machine	1	MIL-S-901 4A mounting	ASTIA-AD-67250	Passed	–

[a]Estimated.

TABLE A-1-22. CONTACTOR

General Description: 1. Type IFN-630 contactor unit, 3-pole, 600-ampere, 440-volt ac, 3-phase, 60-cycle, size 6 contactor.
2. Two temperature-actuated units of the integral bulb type provided with 3/4 in. male pipe fittings for insertion into pipe fittings in service.

Type	Approximate Weight (lb)	Test Environment	Test Machine	Number Tested	Test Specification	Report Designation	Result	Type of Failure
IFN-630	105	1-, 3-, and 5-ft drops	Navy LWHI shock machine	1	MIL-S-901 4A mounting	ASTIA-AD-139914	Passed	–
Bulb type	100[a]	1-, 3-, and 5-ft drops	Navy LWHI shock machine	1	MIL-S-901 4A mounting	ASTIA-AD-20710	Failed 5-ft, rear	b

[a]Estimated.
[b]Bending of the sensitive element of the bulb type temperature switch.

TABLE A-1-23. CONTROLLER

General Description: 1. Two-speed motor controller at 25-hp, 50-amp, 440-volt, 3-phase, 60-cycle ac, 3-pole, double-throw contactor, one relay, 2 overload relays start and stop buttons, speed selector, and control switch.
2. Magnetic, size 1, across the line starter designed for 2- or 3-wire control.
3. Manually-operated, across the line motor starter rated at 7-1/2 hp, 440-volt, 3-phase, 60-cycle ac.

Type	Approximate Weight (lb)	Test Environment	Test Machine	Number Tested	Test Specification	Report Designation	Result	Type of Failure
25 hp	90	1-, 3-, and 5-ft drops	Navy LWHI shock machine	1	MIL-S-901 4A mounting	ASTIA-AD-45071	Passed	–
7-1/2 hp	29	1-, 3-, and 5-ft drops	Navy LWHI shock machine	1	MIL-S-901 4A mounting	ASTIA-AD-67242	Passed	–
Manually operated	12	1-, 3-, and 5-ft drops	Navy LWHI shock machine	1	MIL-S-901 4A mounting	ASTIA-AD-56169	Passed	–

TABLE A-1-24. DIESEL GENERATORS

General Description: 1. Two-cycle, 6-cylinder, diesel engine and a direct driven 60-kw generator.
2. Four-stroke, 1 cylinder, diesel engine and a 2.5-kw, 115-volt ac, 60-cycle generator.
3. An air cooled cast aluminum diesel engine directly connected to a 5-kw, 28-volt dc generator.

Type	Approximate Weight (lb)	Test Environment	Test Machine	Number Tested	Test Specification	Report Designation	Result	Type of Failure
60 kw	3777	1- and 3-ft drops	Navy MWHI shock machine	1	MIL-S-901	ASTIA-AD-204925	Failed 3 ft, top	a
2.5 kw	434	1- and 3-ft drops	Navy MWHI shock machine	1	MIL-S-901	ASTIA-AD-201051	Failed 3 ft, top	b
5 kw	663	1- and 3-ft drops	Navy MWHI shock machine	1	MIL-S-901	ATI-204607	Failed 3 ft, top	Crack in fly-wheel housing

[a]Cracked flywheel housing and severe structural damage to the front support of the engine.
[b]Extreme damage to the mounting pad and front support.

TABLE A-1-25. ENCAPSULATED FITTINGS

General Description: 1. RG 14A/U encapsulated pressure barrier designated for Halibut (SSGN 587).
2. Type 14 AU encapsulated potted fittings for the Halibut (SSGN 587).

Type	Approximate Weight (lb)	Test Environment	Test Machine	Number Tested	Test Specification	Report Designation	Result	Type of Failure
RG 14A U	6	1-, 3-, and 5-ft drops	Navy LWHI shock machine	1	MIL-S-901B 4A mounting	Mare Island NavShipYd 5931-59	Passed	–
14AU	31	1-, 3-, and 5-ft drops	Navy LWHI shock machine	2	MIL-S-901B 4C mounting	Mare Island NavShipYd 4307-59	Passed	–

TABLE A-1-26. FUSE BOX

General Description: A tank indicator fuse box (SSGN 587).

Type	Approximate Weight (lb)	Test Environment	Test Machine	Number Tested	Test Specification	Report Designation	Result	Type of Failure
Tank indicator	30	1-, 3-, and 5-ft drops	Navy LWHI shock machine	1	MIL-S-901B 4A mounting	Mare Island NavShipYd 2715-59	Passed	–

TABLE A-1-27. GEAR MOTOR

General Description: Gear motor for helicopter lift.

Type	Approximate Weight (lb)	Test Environment	Test Machine	Number Tested	Test Specification	Report Designation	Result	Type of Failure
Gear motor	232 motor 100 foundation	1-, 3-, and 5-ft drops	Navy LWHI shock machine	1	MIL-S-901B 4A mounting	Mare Island NavShipYd 0418-60	Passed	–

TABLE A-1-28. MOTORS

General Description: Several reports were reviewed and then grouped into one of two power classes.

0 to 25-hp Class

1. 6-hp, 245-volt dc, Navy A-type motor.
2. 15/10-hp, 440-volt ac motor.
3. 15-hp, 230-volt dc, Navy A service type motor.
4. 17.5-hp, 440-volt ac, air conditioning compressor motor.
5. 25-hp 500-volt dc, hydraulic power plant on SS563 submarine motor.

25 to 50-hp Class

1. 30-hp, 250-volt dc, high pressure air compressor.
2. 30-hp, 355/250-volt dc, high pressure air compressor motor. SSK1, SSK2, SSK3 motors.
3. 55-hp, 500-volt dc motor for a trim pump on SS563 submarines.

Type	Approximate Weight (lb)	Test Environment	Test Machine	Number Tested	Test Specification	Report Designation	Result	Type of Failure
0 to 25 hp	302-625	0.75-, 1.75-, and 1.75-ft drops	Navy MWHI shock machine[a]	5	N.D. 66S3	EES-5B70001 AD200998 EES-5A(2) 51832 AD201042 EES-5UX1609	1 failure	b
26 to 55 hp	1169-1727	1-, 2-, and 2-ft drops	Navy MWHI shock machine[a]	3	N.D. 66S3	AD205025 AD201028 EES-5AA1 X1609	Passed	–

[a]All motor types tested on Navy MWHI shock machines.
[b]The 15-hp motor failed on a 0.75-foot back blow. The brush rigging supports broke loose from the insulating ring on the housing of the motor.

TABLE A-1-29. GENERATORS

General Description: 1. 60-kw, 240/120-volt dc, turbine driven generator.
2. 250-kw ac, 3-phase, 60-cycle ship's service generator with 5.5-kw exciter.

Type	Approximate Weight (lb)	Test Environment	Test Machine	Number Tested	Test Specification	Report Designation	Result	Type of Failure
60 kw	2100	1.25-, 2.25-, and 2.25-ft blows	Navy MWHI shock machine	1	N.D. 6683	EES-C-2526-2	Passed	–
250 kw	3965	1.75-, 2.75-, and 2.75-ft blows	Navy MWHI shock machine	1	N.D. 6683	EES-C-2593	Failed	End bracket fractured

TABLE A-1-30. MOTOR GENERATORS

General Description: 1. 1.4-kw, 56-volt dc generator and 3-hp, 440-volt ac motor for battery charging in ships service dial telephone equipment.
2. 2.8-kw, 56-volt dc generator and 5-hp, 440-volt ac motor also for ships service dial telephone equipment.
3. 25-kva, 120-volt generator and 30-hp, 220-volt battery.

Type	Approximate Weight (lb)	Test Environment	Test Machine	Number Tested	Test Specification	Report Designation	Result	Type of Failure
1.4 kw	315	0.75-, 1.75-, and 1.75-ft blows	Navy MWHI shock machine	1	N.D. 6683	EES-5QX1609	Passed	–
2.8 kw	501	0.75-, 1.75-, and 1.75-ft blows	Navy MWHI shock machine	1	N.D. 6683	EES-5KX1603	Passed	–
25 kva	1320	1-, 2-, and 2-ft blows	Navy MWHI shock machine	1	MIL-S-901	EES-5AKX-1609 AD201040	Passed	–

TABLE A-1-31. PANELS

General Description: 1. Air sampler alarm system relay panel.
2. Type B-52 alarm panel.
3. Aft remote control panel for main hydraulic pumps and accum Nos. 1 and 2.

Type	Approximate Weight (lb)	Test Environment	Test Machine	Number Tested	Test Specification	Report Designation	Result	Type of Failure
B-52 air sampler, alarm sampler	10	1-, 3-, and 5-ft blows	Navy LWHI shock machine	1	MIL-S-901B 4A mounting	Mare Island NavShipYd 2751-59	Passed	–
Type B-52 alarm	11	1-, 3-, and 5-ft blows	Navy LWHI shock machine	1	MIL-S-901B 4A mounting	ASTIA-AD-206035	Passed	–
Control	120	1-, 3-, and 5-ft blows	Navy LWHI shock machine	1	MIL-S-901B 4A mounting	Mare Island NavShipYd 2750-59	Passed	–

TABLE A-1-32. PLUGS AND RECEPTACLES

General Description: 1. Three-contact plugs with 3-contact receptacles with triple conductor cable and rated at 15 amps.
2. Three 10-amp plugs and three 10-amp receptacles. Receptacles are in water tight drawn brass enclosure.
3. Two 40-amp plugs and receptacles in water tight drawn brass enclosures.

Type	Approximate Weight (lb)	Test Environment	Test Machine	Number Tested	Test Specification	Report Designation	Result	Type of Failure
15 amp	2[a]	1-, 3-, and 5-ft drops	Navy LWHI shock machine	1	MIL-S-901B 4A mounting	ASTIA-AD-86539	Passed	–
10 amp	2.25	1-, 3-, and 5-ft drops	Navy LWHI shock machine	3	MIL-S-901B 4A mounting	ASTIA-AD-86534	Passed	–
40 amp	3.6	1- and 3-ft drops	Navy LWHI shock machine	2	MIL-S-901B 4A mounting	ASTIA-AD-86534	Failed 3 ft, back	b

[a]Estimated.
[b]The plug became wedged against the receptacle in such a way that removal was very difficult.

TABLE A-1-33. RECTIFIER

General Description: Type A selenium rectifier panel for battery charging, designed for 50°C ambient temperature and rated at 440-volt, 3-phase, 60-cycle input with 2- to 6-amp dc output.

Type	Approximate Weight (lb)	Test Environment	Test Machine	Number Tested	Test Specification	Report Designation	Result	Type of Failure
Selenium rectifier	180	1-, 3-, and 5-ft drops	Navy LWHI shock machine	1	MIL-R-15736 (bulkhead mounted)	ASTIA-AD-37551	Passed	–

TABLE A-1-34. RELAYS

General Description: Many reports on relays were reviewed. The following is only a brief description:
1. The type IAC time overcurrent relay consists of an induction disk operating mechanism, a set of single-pole contacts, and a dust tight sheet steel enclosure. It is used to trip a circuit breaker when over-current conditions occur.
2. The auxiliary relays consisted of a solenoid operating magnet and plunger which is cross-connected to the contact assemblies.
3. Five bulletin 130 auxiliary control relays, from 150- to 440-volt ac coil voltage.

Type	Approximate Weight (lb)	Test Environment	Test Machine	Number Tested	Test Specification	Report Designation	Result	Type of Failure
IAC	11.5	1-, 3-, and 5-ft drops	Navy LWHI shock machine	1	MIL-S-901 6D mounting	ASTIA-AD-111803	Passed	–
Auxiliary	4[a]	1-, 3-, and 5-ft drops	Navy LWHI shock machine	8	MIL-S-901 6E mounting	ASTIA-AD-139798	2 failed (back and top, 3 and 5 ft)	Normally closed contacts opened
Control	1.3 to 4.3	1-, 3-, and 5-ft drops	Navy LWHI shock machine	5	6683 6E mounting	Material Lab NY NavShipYd 5225-1	Failed (all blows)	b

[a]Estimated.
[b]Malfunction of the electrical contacts.

TABLE A-1-35. RESISTORS

General Description: 1. Fixed, accurate, wire wound resistors from 10 to 3750 kilohms.
2. Variable subminiature resistors from 100 to 5,000,000 ohms.

Type	Approximate Weight (lb)	Test Environment	Test Machine	Number Tested	Test Specification	Report Designation	Result	Type of Failure
Wire wound	3[a]	30 impacts at 50 g with 0.012 sec duration	Navy LWHI shock machine	1152	Battelle Memorial Institute	Signal Corps Project 2006-A	Passed	—
Miniature	2[a]	30 impacts at 50 g with 0.012 sec duration	Navy LWHI shock machine	40	Battelle Memorial Institute	Signal Corps Project 2006-A	Passed	—

[a]Estimated.

TABLE A-1-36. SWITCHBOARDS

General Description: 1. The magazine-sprinkling-alarm switchboard is a monitoring unit that visually and audibly indicates conditions in remote compartments as reported by sensing elements.
2. The magazine sprinkling alarm switchboard is designed to automatically monitor the operation of the firefighting sprinkling system installed in the ship's magazines (10 lines).

Type	Approximate Weight (lb)	Test Environment	Test Machine	Number Tested	Test Specification	Report Designation	Result	Type of Failure
Alarm magazine sprinkling	100[a]	1-, 3-, and 5-ft drops	Navy MWHI shock machine	1	MIL-A-17196A	ASTIA-AD-77815	Passed	—
10-line alarm	109	1-, 3-, and 5-ft drops	Navy MWHI shock machine	2	MIL-S-901 4A mounting[b]	ASTIA-AD-23734	Failed	False alarms

[a]Estimated.
[b]As detected in photograph.

TABLE A-1-37. LIMIT SWITCHES

General Description: 1. Micro limit switch type ML-X14928 for the Guided Missile Launching System Mk 9.
2. The D1450 trackway limit switch is rated to carry 20 amps at 440 volts ac or 230 volts dc.
3. Limit Switches Type A Mk 6.

Type	Approximate Weight (lb)	Test Environment	Test Machine	Number Tested	Test Specification	Report Designation	Result	Type of Failure
Micro switch	20[a]	1-, 3-, and 5-ft blows	Navy LWHI shock machine	5	MIL-S-901B 6E mounting	ASTIA-AD-48787	Passed	—
Trackway	22	1-, 3-, and 5-ft blows	Navy LWHI shock machine	3	MIL-S-901 4A mounting	ASTIA-AD-145797	Passed	—
Mk 6	20[a]	1-, 3-, and 5-ft blows	Navy LWHI shock machine	6	MIL-S-901B 6E mounting	ASTIA-AD-129696	Failed	Fluttering and opening of the normally-closed circuit

[a]Estimated.

TABLE A-1-38. ROTARY SWITCHES

General Description: 1. Six miniature rotary switches. Three contained bushings and shaft seals and the other three were equipped with boots to provide water tightness.
2. Three Type 2JL5 and three Type 2JL10 rotary selector switches.
3. Three each of the following: S-2JF1, S2JF3, and S2JF5 rotary switches.
4. One each of rotary switches, Type S2JR10 and S2JR25.

Type	Approximate Weight (lb)	Test Environment	Test Machine	Number Tested	Test Specification	Report Designation	Result	Type of Failure
Miniature	5	1-, 3-, and 5-ft blows	Navy LWHI shock machine	6	MIL-S-901B 4A mounting	ASTIA-AD-214226	Failed	a
Selector 2J5L5 and 2JL10	0.5 1.125	1-, 3-, and 5-ft blows	Navy LWHI shock machine	6	MIL-S-901B 6D mounting	ASTIA-AD-207729	3 Type-2J5L5 failed (5 ft, back) 3 Type-2JL10 passed	Shaft ejected completely from the switch
S2JF1 S2JF3 S2JF5	3.3[b] 3.75[b] 4.25[b]	1-, 3-, and 5-ft blows	Navy LWHI shock machine	9	MIL-S-901B 6D mounting	ASTIA-AD-207741	Failed (5 ft, back)	Distortion of components
S2JR10 S2JR25	2.3 4.6	1-, 3-, and 5-ft blows	Navy LWHI shock machine	2	MIL-S-901B 7C mounting	ASTIA-AD-207728	Passed	—

[a]The shaft was ejected from the sample with the bushing and shaft seal.

[b]Weight in ounces.

TABLE A-1-39. ROTARY SNAP SWITCHES

General Description:
1. Seven 60-amp, 450-volt ac, 250-volt dc, Type 6SR3A1 base-mounted, rotary snap switches.
2. Seven 60-amp, 450-volt ac, 250-volt dc, Type 6SR3A1 panel-mounted, rotary snap switches.
3. Seven 10-amp, 120-volt ac, rotary snap switch Type 1SR3A1, TPST.
4. Seven 30-amp, 500-volt ac, 250-volt dc, Type 3SR3B4 base-mounted, rotary snap switches.
5. Seven 10-amp, 120-volt ac, Type 1SR3B1, TPDT, front-mounted for panels with handles.
6. Three each of Types 20SR3B1 and 20SR6F1, 200-amp, rotary snap switches that are panel mounted.
7. One Type 91740-A5 switch and one Type 91741-A5 switch. Both are miniature rotary switches for interior communication equipment.
8. Models CA-54, CA-55 and CA-56 rotary solenoid switches for high altitude use.
9. Ten rotary switches of non-magnetic construction were reviewed.
10. Two Type A-384-5WA rotary selector switches without a shaft "O" ring groove.
11. Four 8-position, 25-section, multipole rotary switches.

Type	Approximate Weight (lb)	Test Environment	Test Machine	Number Tested	Test Specification	Report Designation	Result	Type of Failure
6SR3A1 (base mounted)	7.6	1-, 3-, and 5-ft blows	Navy LWHI shock machine	7	MIL-S-901B 4A mounting	ASTIA-AD-129390	Passed	—
6SR3A1 (panel mounted)	5.9	1-, 3-, and 5-ft blows	Navy LWHI shock machine	7	MIL-S-901B 4A mounting	ASTIA-AD-129389	Passed	—
1SR3A1 1SR3B1	0.75 0.5	1-, 3-, and 5-ft blows	Navy LWHI shock machine	7	MIL-S-901B 4A mounting	ASTIA-AD-129388	Passed	—
3SR3B4	2.1	1-, 3-, and 5-ft blows	Navy LWHI shock machine	7	MIL-S-901B 4A mounting	ASTIA-AD-129386	Passed	—
1SR3B1	—	1-, 3-, and 5-ft blows	Navy LWHI shock machine	7	MIL-S-901B 4A mounting	ASTIA-AD-206102	Passed	—
20SR3B1 20SR6F1	1[a]	1-, 3-, and 5-ft blows	Navy LWHI shock machine	2 1	MIL-S-901 4C mounting	ETL-1367 Electrical Testing Lab Portsmouth, N.H.	Passed	—
91740-A5 91741-A5	1.5	1-, 3-, and 5-ft blows	Navy LWHI shock machine	2	MIL-S-901B 4A mounting	ASTIA-AD-214227	Failed	Porcelain spacer fractured
CA-54 CA-55 CA-56	2 2 5	20 g	Navy LWHI shock machine	3	JAN-S-44	ASTIA-AD-19174	Passed	—
Non-magnetic	7[a]	1-, 3-, and 5-ft blows	Navy LWHI shock machine	10	MIL-S-901B 6D mounting	ASTIA-AD-209987	Failed	Mounting plate buckled
A-384-5WA	7[a]	1-, 3-, and 5-ft blows	Navy LWHI shock machine	2	MIL-S-901B 6D mounting	ATI-195747	Passed	—
Multipole	7[a]	1-, 3-, and 5-ft blows	Navy LWHI shock machine	4	MIL-S-901B 6D mounting	ATI-210278	Passed	—

[a]Estimated.

TABLE A-1-40. TOGGLE SWITCHES

General Description: 1. Eighteen miniature toggle switches, Type T2104, are miniaturized two-circuit switches designed for use in applications where panel space is at a premium.
2. Environmental-proof multipole, multiposition, toggle switch.
3. Twenty different types of toggle switches.

Type	Approximate Weight (lb)	Test Environment	Test Machine	Number Tested	Test Specification	Report Designation	Result	Type of Failure
T2104	0.5[a]	75 g	Navy LWHI shock machine	18	JAN-S-23	ASTIA-AD-111723	Passed	–
Environmental toggle switch	0.24	50 g for 0.5 to 9.7 sec duration	Navy LWHI shock machine	1	AER-EL-62 XEL-126 AN-5-200	ATI-96739	Failed	Momentary opening of contacts
Toggle switches	0.5 ea.	1-, 3-, and 5-ft blows	Navy LWHI shock machine	20	MIL-S-901B 6D mounting	ASTIA-AD-214225	Failed	–

[a]Estimated.

TABLE A-1-41. MISCELLANEOUS SWITCHES

General Description:
1. Five temperature-operated master switches and six pressure-operated automatic master switches.
2. One stepping switch, 20-volt ac, Type 58-2.
3. Eight switches each of Types D8-4 (15-amp at 125/250-volt ac), S3-204, S1-4 (10 amp at 125/250-volt ac), E3-4 (2.5 amp and 5 amp at 125/250-volt ac, and 7 Type S2-104 switches (10 amp at 125/250-volt ac).
4. Forty hinge roller leaf type sensitive switches SS07A10, SS07A20, SS07B40, and SS07B7Q.
5. Two sample pressure proof 1- to 7-Mc switches.

Type	Approximate Weight (lb)	Test Environment	Test Machine	Number Tested	Test Specification	Report Designation	Result	Type of Failure
Temperature and pressure operated switches	2.2	1-, 3-, and 5-ft blows	Navy LWHI shock machine	11	MIL-S-901 6E mounting	ASTIA-AD-36805	Passed	—
Stepping switch	5[a]	1-, 3-, and 5-ft blows	Navy LWHI shock machine	1	MIL-S-901B 6E mounting	ASTIA-AD-59337	Failed	Cam forced out of position
S3-204 S2-104	1[a] each	1-, 3-, and 5-ft blows	Navy LWHI shock machine	8 7	MIL-S-901B 6E mounting	ASTIA-AD-48789	Failed	b
Roller leaf	3 each	1-, 3-, and 5-ft blows	Navy LWHI shock machine	32	MIL-S-901A 6E mounting	ASTIA-AD-41605	Passed	—
Pressure-proof	9.8	1-, 3-, and 5-ft blows	Navy LWHI shock machine	2	MIL-S-901B 4A mounting	ASTIA-AD-145281	Failed	Diaphragm revealed leakage

[a]Estimated.
[b]Transfer contacts of each switch made contact with the normally-open contact of its respective switch.

TABLE A-1-42. TEMPERATURE ELEMENTS

General Description: The elements were the disk, bead, or rod shape type.

Type	Approximate Weight (lb)	Test Environment	Test Machine	Number Tested	Test Specification	Report Designation	Result	Type of Failure
Elements	4[a]	30 impacts at 50 g for 8 to 12 ms duration	Navy LWHI shock machine	102	PR and C No. 59-FLS/D-3431	Signal Corps Project 2006-A	Passed	—

[a]Estimated.

TABLE A-1-43. VOLTAGE SENSITIVE ELEMENTS

General Description: Eighty specimens were tested. The specimens differed from one another in material (plastic, glass, or metal), shape (tubular or disk), and mounting (by soldering, clamping, or bolting).

Type	Approximate Weight (lb)	Test Environment	Test Machine	Number Tested	Test Specification	Report Designation	Result	Type of Failure
Elements	Below 1[a]	30 impacts at 50 g; each impact applied for 8 to 12 ms	Navy LWHI shock machine	80	Battelle Memorial Institute	Signal Corps Project No. 2006-A	Passed	—

[a]Estimated.

A-2. Transient Shock Tests of Commercial Communication Equipment.

A-2-1.

a. Representative operating communications system for testing. The communications subsystem major components were shock- and vibration-tested rack by rack during simulated operation. The equipment considered for test must survive a nuclear engagement. The equipment constituted standard GTE Lenkurt, Western Electric, and Government furnished assemblies and supplies comprising a fully equipped terminal. The subsystem included the following nine test articles:

(1) Lenkurt 46A Multiplex Equipment consisting of 1 channel bank, 3 groups and 2 supergroups (2 and 3) and associated supplies. Two 11½ ft standard racks (tested separately).

(2) Lenkurt 46C End Terminal Equipment consisting of two (redundant) fixed-gain transmitting and two (redundant) regulation controlled receiving line amplifiers; one 11½ ft standard rack.

(3) One Lenkurt 46C Dependent Repeater consisting of one transmitting and receiving line amplifier, two cable termination stubs, and a shock isolation system. The redundant cabinet was weighted to simulate mass of operating components.

(4) Western Electric Equipment consisting of one 203A Data Set, one 303B/LWM-4 Data Set/Wideband Modem, and one 303C/LWM-6 Data Set/Wideband Modem; one 11½ ft rack bulb type and retest of same equipment installed in a channel rack.

(5) Western Electric 111A Power Plant Equipment with distribution facilities and supplies; three 7-ft racks tested together as a unit.

(6) Cryptographic Equipment consisting of 3 KG-34 units; government furnished—one 7-ft rack, CY 597-G equipment cabinet.

(7) Western Electric Inverter, one 7-ft standard rack.

(8) Western Electric 2-tier Battery Stand e/w 24 batteries, one unit.

b. The Lenkurt 46A multiplex, 46C end terminal, 46C repeater and Western Electric 111A power plant equipment constitute a transistorized, fully alarmed transmission system with sufficient options to permit total redundancy of main active components.

A-2-2. Protective facility criteria environment. The overall transient shock environment imposed on the protective facility by airblast and ground shock is given in figure A-2-1 for shock spectra and figure A-2-2 for acceleration time-history waveforms. The waveforms were synthesized to match the shock spectra in accordance with Yang (1971).

A-2-3. Shock isolation systems. Telephone equipment was mounted upon shock isolated platforms. These shock isolated platforms ranged in size from 11½ ft by 14 ft to 27 ft by 39½ ft. Isolators in all cases were pendulum mounted for horizontal motion and pneumatic for vertical motion. Number of isolators ranged from 4 to 8 for each platform. Isolated platform rigid body design frequencies were 0.2 Hz horizontal and 0.75 Hz vertical.

A-2-4. Transient shock environments. The equipment transient shock environments of facility motions (fig. A-2-1) transmitted through the shock isolation/platform systems to equipment locations were determined by finite element modeling of the shock isolation systems. Response motions calculated used the facility acceleration time histories given in figure A-2-2 as input functions. A typical model of a platform, supporting superstructure, equipment, and isolators is shown in figure A-2-3.

A-2-5. Environment on shock isolated platforms. The shock spectra for the equipment test environment are shown in figures A-2-4 and A-2-5. The levels shown in the figures are representative for the various platforms evaluated as well as the location of equipment on the platforms. In general, the levels selected are conservative. The 100 percent level or reference level is the best estimate of the environment (including uncertainties) at the attachment points of the equipment. The 200 percent level represents the worse case estimate but more importantly serves to provide hardness information of equipment. The 50 percent level may be regarded as the optimistic estimate of the environment. Advantageously, for this program, the control system for the test machine required a series of successive iterations due to nonlinearities in the test machine/system. This control limitation led in part to the typical test-level sequence for each axis in table A-2-1. Three tests at the 100 percent level were specified for test article qualifications.

A-2-6. Time histories. Acceleration time-history waveforms were generated for the vibration test machine by damped sine waves but were consistent in effect to Yang (1971) Typical data obtained during tests at the 100 percent level are shown in figure A-2-6.

A-2-7. Results. Summaries of test results for each equipment tested are given in tables A-2-2 and A-2-3. It is to be noted that the 46C dependent terminal is directly mounted on two coil spring isolators (with friction dampers) in a manhole enclosure. The following tables and data plots are all taken from the first reference under paragraph A-2-8 below.

A-2-8. References and credits.

Safford, F. B. *Environmental Test and Hardness Verification of SAFCA Equipment, Task 5 Hardness/Vulnerability Assessment*, R-7237-3365. El Segundo, CA: Agbabian Assoc., Jul 1974.

Safford, F. B. and Tuttle, R. J. *Transient Shock Fragility of Commercial Communications Equipment*, 740801. Warrendale, PA: Soc. of Auto Engrs., Oct 1974.

Interim Test Report on Shock Test Program for IIIA Power Equipment, No. 58720-06. Huntsville, AL: Wyle Laboratories, Oct 1973.

Interim Test Report on Shock Program for Inverter Rack, No. 5820-24. Huntsville, AL: Wyle Laboratories, Oct 1973.

Interim Test Report on Shock Test Program for 203/303 Data Equipment, No. 58720-13-1. Huntsville, AL: Wyle Laboratories, Oct 1973.

Interim Test Report on Shock Test Program for Western Electric Battery Stand, No. 58720-23. Huntsville, AL: Wyle Laboratories, Nov 1973.

Interim Test Report on Shock Test Program for LENKURT 46A Multiplex Equipment Unit 1, No. 58720-09. Huntsville, AL: Wyle Laboratories, Jan 1974.

Interim Test Report on Shock Test Program for LENKURT 46A Multiplex Equipment Unit 2, No. 58720-10. Huntsville, AL: Wyle Laboratories, Jan 1974.

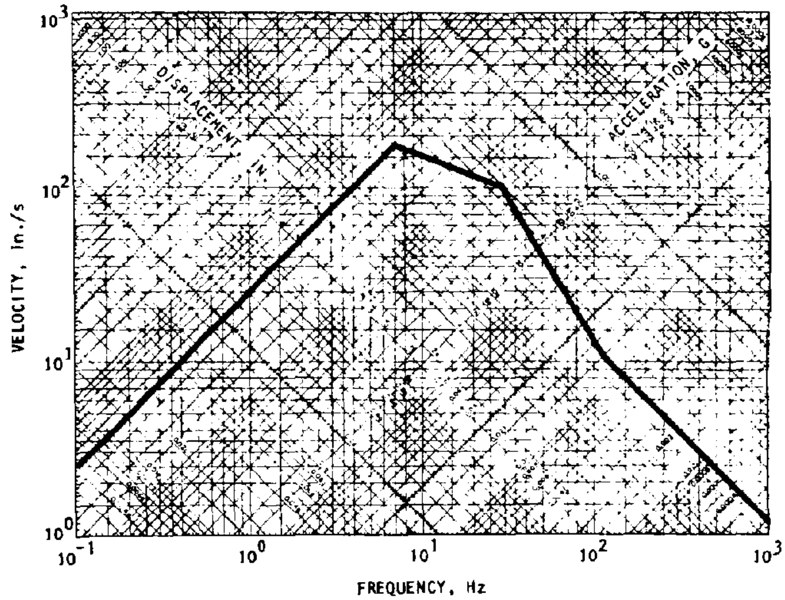
Interim Test Report on Shock Test Program for KG-34 Crypto Equipment, No. 58720-16. Huntsville, AL: Wyle Laboratories, Nov 1973.

Interim Test Report on Shock Test Program for LENKURT 46C Terminal Equipment, No. 5870-15. Huntsville, AL: Wyle Laboratories, Jan 1974.

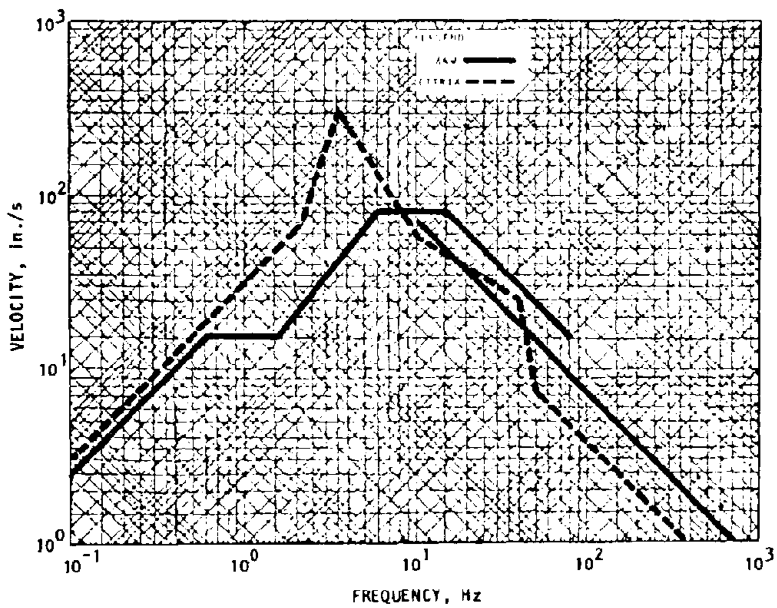
Interim Test Report on Shock Test Program for WECO 203/303 Data Equipment (Channel Rack), No. 58720-13-2. Huntsville, AL: Wyle Laboratories, Jan 1974.

Interim Test Report on Shock Test Program for 46C Repeater, No. 58720-17. Huntsville, AL: Wyle Laboratories, Mar 1974.

Final Test Report on Shock Test Program for SAFEGUARD Communications Agency, No. 58720-25. Huntsville, AL: Wyle Laboratories, Mar 1974.



(a) Test criteria, vertical axis



(b) Test criteria, horizontal axis

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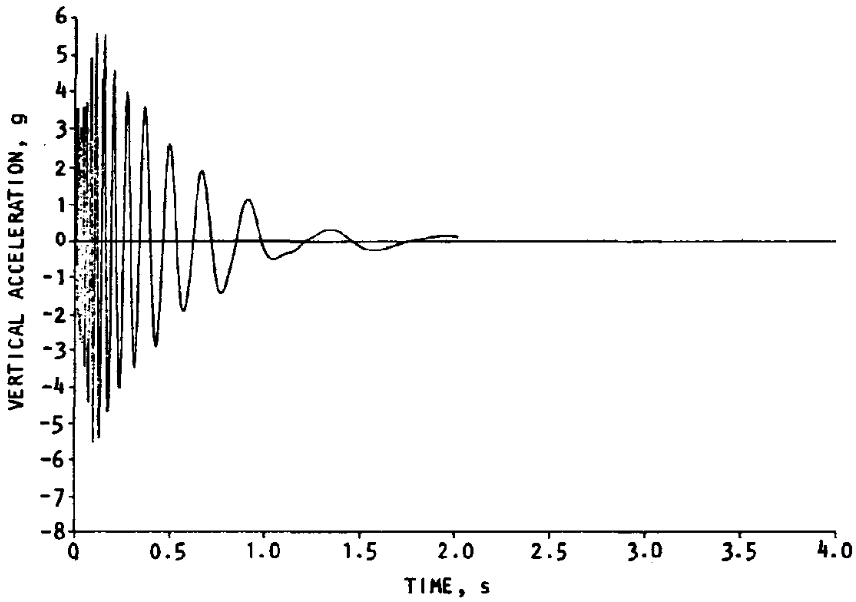
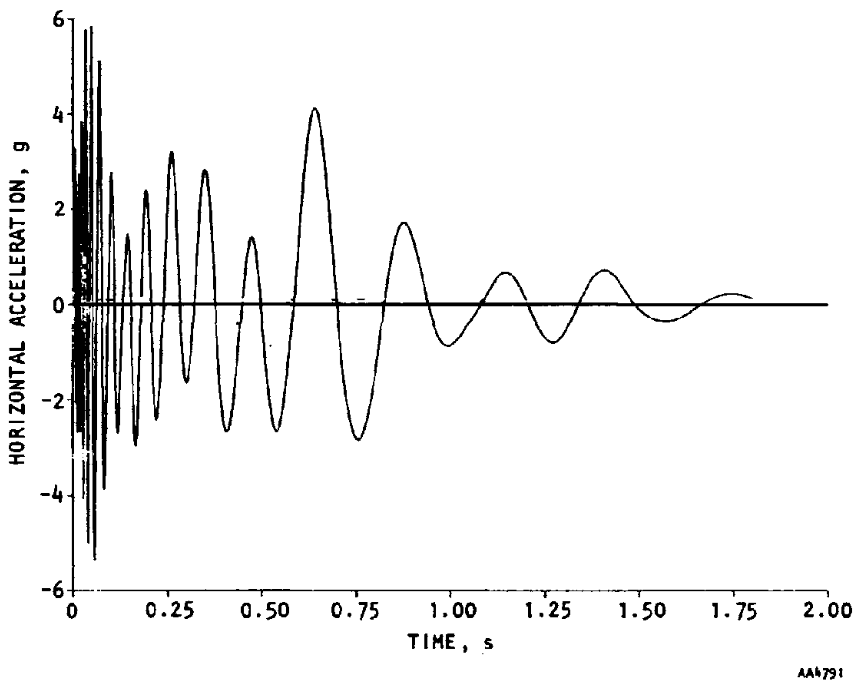
FIGURE A-2-1. VERTICAL AND HORIZONTAL SHOCK SPECTRA FOR PROTECTIVE FACILITY



(a) Vertical acceleration



(b) Horizontal acceleration

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FIGURE A-2-2. ACCELERATION WAVEFORMS

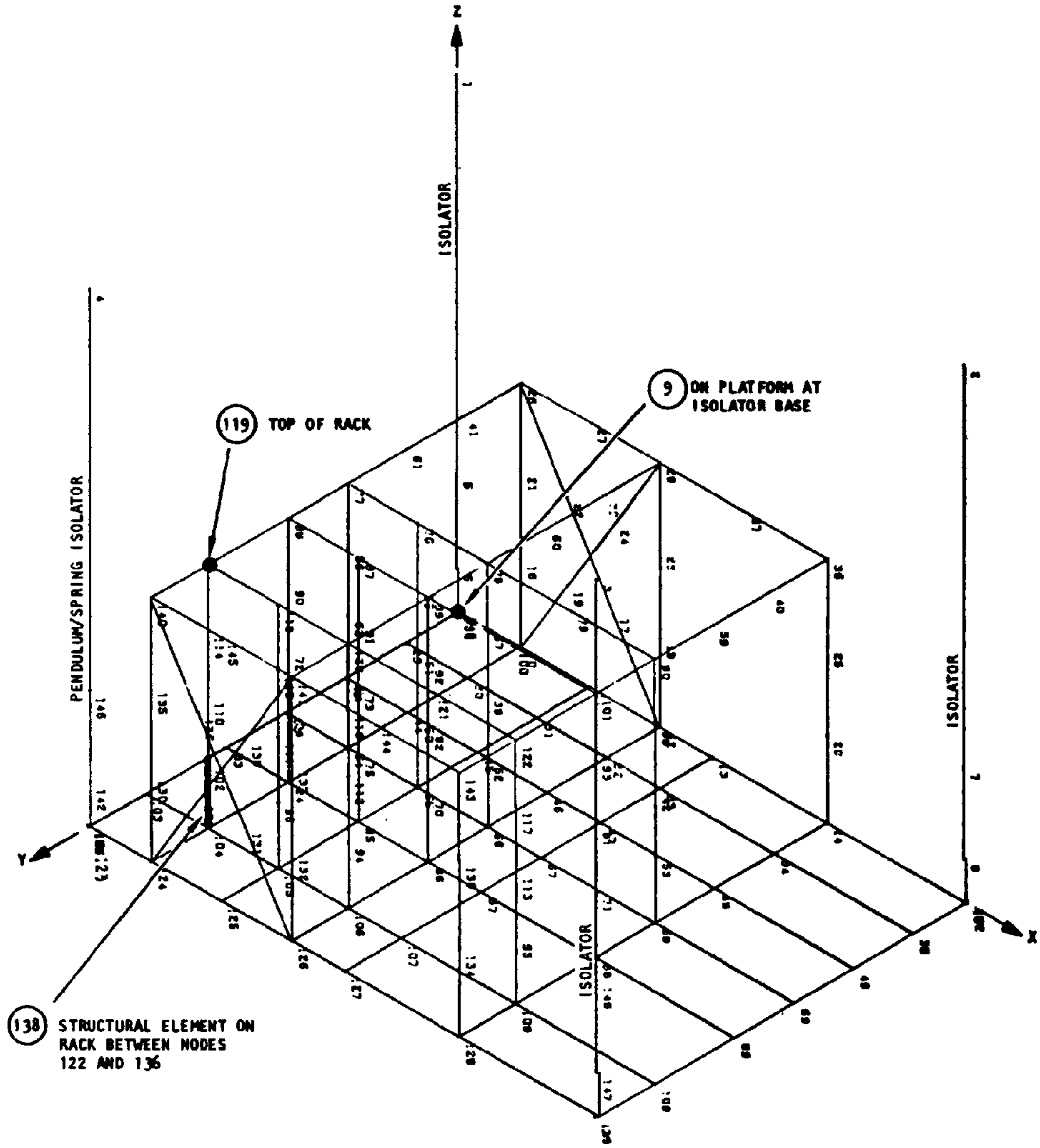

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FIGURE A-2-3. MODEL FOR PLATFORM

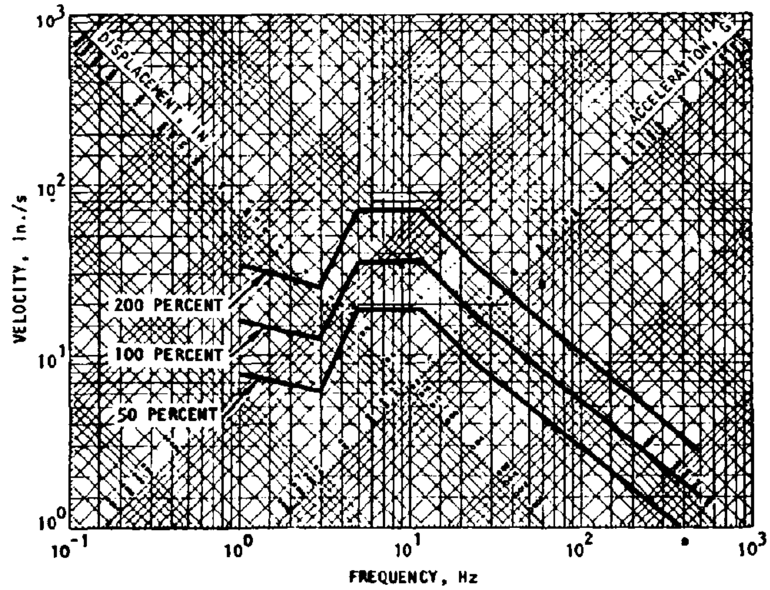

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FIGURE A-2-4. VERTICAL UNDAMPED SHOCK SPECTRA, ALL EQUIPMENT RACKS AND BATTERY RACK

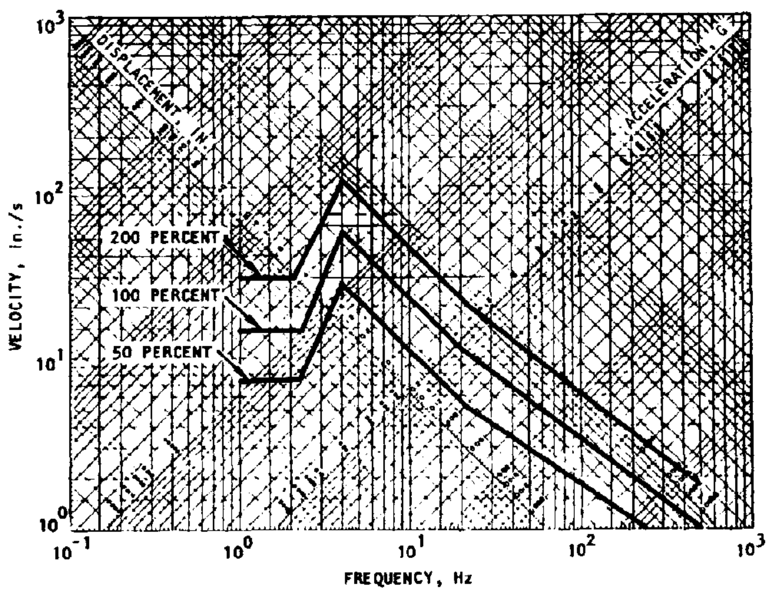

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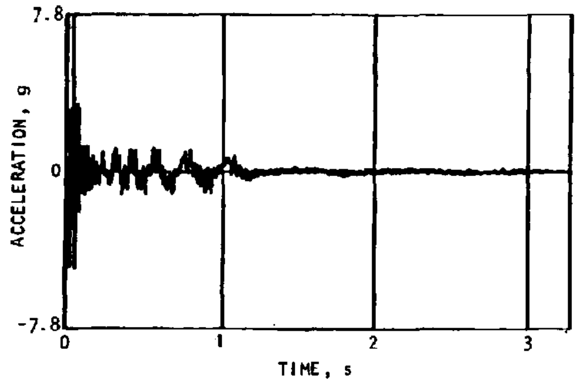
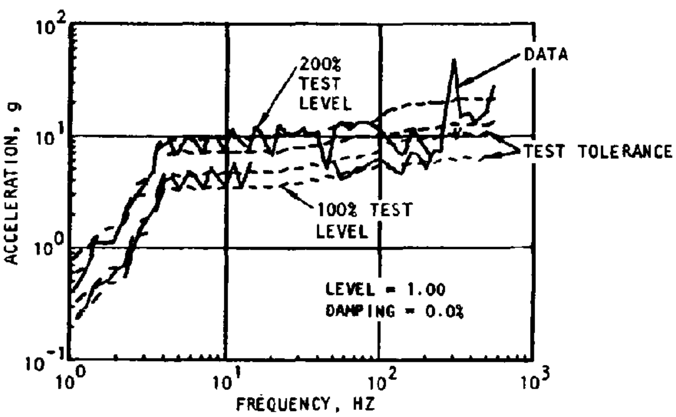
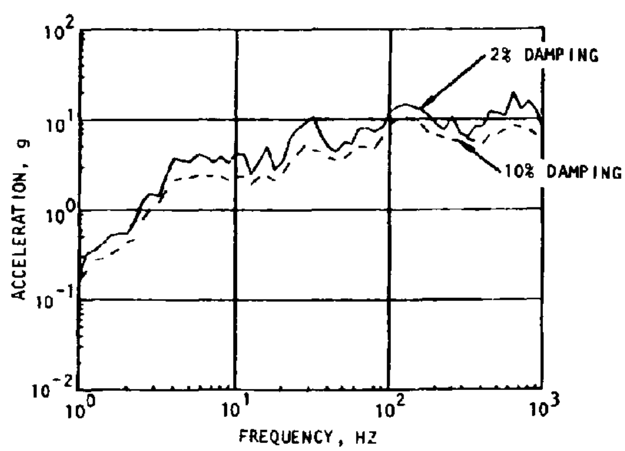
FIGURE A-2-5. HORIZONTAL UNDAMPED SHOCK SPECTRA, ALL EQUIPMENT RACKS AND BATTERY RACK



(a) Test machine input, acceleration (g's) versus time—
100 percent level



(b) Test machine input, shock spectra (g's)



(c) Motion at base of test article (g's)—
100 percent test level

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FIGURE A-2-6. TYPICAL HORIZONTAL TEST MACHINE INPUT AND MOTION AT BASE OF TEST ARTICLE (46A UNIT 1 MULTIPLEX)

TABLE A-2-1. ITERATIVE TEST LEVEL SEQUENCE FOR EACH AXIS

Iteration Number	Test Level, %	Number of Tests
1	25	1
2-4	50	3
5	75	1
6-8	100	3
9	150	1
10	200	1

TABLE A-2-2. QUALIFICATION (100 PERCENT) LEVEL TEST SUMMARY
(see figs. 4-10a and 4-10b) (Safford-Tuttle, 1974)

Equipment	Failure/Malfunction/Degradation				Test Level Intensity, Percent	Corrective Action	Results
	Operational Effect	Mode	Mechanism	Cause			
IIIA Power Equipment	Degradation (reduced capacity)	Fuse holder fell off	Low spring tension on connectors	Design	100	Fuse panel restrained	Hardened to 100%+
	None	Voltmeter panel fell off	Low friction restraint	Design	100	Panel restraints or latches	Hardened to 100%+
		Doors swing open		Design	50	Door latches	Hardened to 100%+
Inverter	None	None	--	--	100	--	Passed qualification level 100%
Battery Rack	None	None	--	--	100	--	Passed qualification level 100%
Crypto	None	None	--	--	100	--	Passed qualification level 100%
203/303 Data Equipment	None	None	--	--	100	--	Passed qualification level 100%
46A Multiplex Unit No. 1	Degradation (reduced capacity)	4 kHz Generator module disengaged	Low spring tension on connector	Design	50	Module restraint brackets	Hardened to 100%+
46A Multiplex Unit No. 2	Degradation (high error counts in data stream)	Intermittent shorts	Loose wire	Quality control	100	Discovered and corrected at 200% test	See fragility test
46C Terminal Equipment	Malfunction (error counts and interruptions, 1 kHz tone)	Line input unit, loose jumper assy	Low spring tension and friction, connector jumper assy	Quality control or design	50	Split end connectors spring force increased	Hardened to 100%+
46C Dependent Terminal	Degradation (minor effect on isolator)	One friction damper failed to operate	Isolator damper guide pin fell out	Quality control	100	Reassembled damper	Passed qualification level 100%

TABLE A-2-3. FRAGILITY (200 PERCENT) LEVEL TEST SUMMARY (Safford-Tuttle, 1974)
(two times magnitude level of figs. 4-10a and 4-10b)

Equipment	Failure/Malfunction/Degradation				Test Level Intensity, Percent	Corrective Action	Results
	Operational Effect	Mode	Mechanism	Cause			
111A Power Equipment	Degradation (reduced capacity)	Fuse Holder Disengaged	Low Spring Tension and Friction of Connector	Design	200	Fuse Panel Restrained	Hardened to 200% +
Inverter	None	1/4" Bolt Sheared	Cumulative Fatigue	Test Dependent	150	Bolt Replaced	Passed Fragility Level 200%
	Failure	Short Circuit	Loose Bolt on Capacitor	Quality Control	200	Capacitor Remounted	Passed Fragility Level 200%
Battery Rack	None	None	--	--	• 200 Vertical • 150 X- and Y-Axis	--	• Passed Fragility Level 200% Vertical • Passed Level 150% Horizontal Axes
Crypto	None	None	--	--	--	--	200% Level
203/303 Data Equipment	Failure (Data Signals Lost)	Power Supply Fell Out of Rack	Weak Mounting Bracket	Design	X-Axis 200 and Y-Axis 200	Support Bracket Installed	Hardened to 200% +
46A Multiplex Unit No. 1	Failure (Data Signals Lost)	Plug-in Modules Disconnected	Low Spring Tension on Connector	Design	150	Restraint Bars Added	Hardened to 200% +
	Degradation (Slight Signal Interruption)	Relay Contact Chatter	Armature Dynamics	Design	150	None	Passed Fragility Level 200%
46A Multiplex Unit No. 2	Failure (Data Signals Lost and 1kHz Tone Interruptions)	Fuse Operated, Lost Power to Capacitor Bank	Extraneous Loose Wire, Electrical Short	Quality Control	200	Removed Loose Wire	Passed Fragility Level 200%
46C Terminal Equipment	None	--	--	--	200	See Qualification Test	Passed Fragility Level 200%
46C Dependent Repeater	None	--	--	--			• 150% X and Y Axis • 200% Vertical Axis

A-3. Transient Shock Tests of Commercial Electric Power and Conditioning Equipment.

A-3-1. Transient tests performed on commercial switchgear, voltage regulators, circuit breakers, transformers, and miscellaneous air and water conditioning equipment employed either single axis or biaxial vibration machines. On single-axis machines, each orthogonal axis of the equipment was tested sequentially. On biaxial machines, a vertical axis and a horizontal axis were tested simultaneously, followed by a second test in the vertical and the other horizontal axis. Type of test employed is noted for each equipment tested.

A-3-2. Duration of the transient tests ranged from about 1 to 2 sec. Acceleration-time history input motions were synthesized by either an assemblage of decaying sinusoids (Brust, 1961), or by an assemblage of sine beats (Yang-Saffell, 1972). Shock spectra associated with each equipment are specified in the following subappendixes together with test results and document references.

Controllers and Switchgear	*Division*
Motor Control Center	A-3A
Generator Control Panel	A-3B
Electric Motor-Generator Control Center	A-3C
Electric Motor Control Center	A-3D
Air Compressor Control Panel and Drive Motor	A-3E

Controllers and Switchgear	*Division*
Motor Control Center	A-3F
Electric Motor Control Center	A-3G
Circuit Breaker	A-3H
Circuit Breaker Assembly	A-3I
Voltage Regulator and Control Center	A-3J
Substations	
Unit Substation (Switch/Transformer)	A-3K
Unit Substation, Transformer	A-3L
Transformer	A-3M
Electrical Components	
Diesel Engine Components	A-3N
Motor Starter and Surge Pak	A-30
Monitor and Control Components	A-3P
Relay Panel	A-3Q
Gas Turbine	
Gas Turbine/AC Generator	A-3R
Conditioning Equipment	
Chiller	A-3S
Cooling Coils	A-3T
Air Handling Unit	A-3U
Heat Sensor	A-3V
Air Filter	A-3W
Piping	
Piping Segments	A-3X
Piping Segments	A-3Y
Miscellaneous Components	
Monitor and Control Duct Mounted Equipment	A-3Z
Shutdown Switch	A-3AA
Pump, Motor, and Valve	A-3BB
Pressure Control Valve	A-3CC

A-3A. MOTOR CONTROL CENTER

1. Motor control center.

a. The specimen was a Westinghouse, Type W, Motor Control Center. Unit's dimensions are approximately 35 in. wide, 15 in. deep, 90 in. high, and weight is about 550 lb. It was provided with:

b. Four NEMA standard full voltage, nonreversing circuit breaker-type combination motor starters (1 size 1, 1 size 2, 1 size 3, and 1 size 4).

c. One 12 circuit 120/208 volt lighting panel with a 100 ampere main circuit breaker.

2. Test.

a. *Shock spectra*.

Horizontal	Figure A-3A-1
Vertical	Figure A-3A-2

b. *Single axis tests*.

3. Results.

a. *Vertical axis*. Relay chatter, breaker opened.

b. *Horizontal axis*. Cabinet twisted, sheet metal sites distorted, some bottom mounting bolts sheared and back panel bolts sheared.

4. References and credits.

The Ralph M. Parsons Co. (Parsons). *Nike-X Data Report: Shock Tests of Six Selected Items of Equipment*, NX-SE-165. Los Angeles, CA: Parsons, Feb 1969.

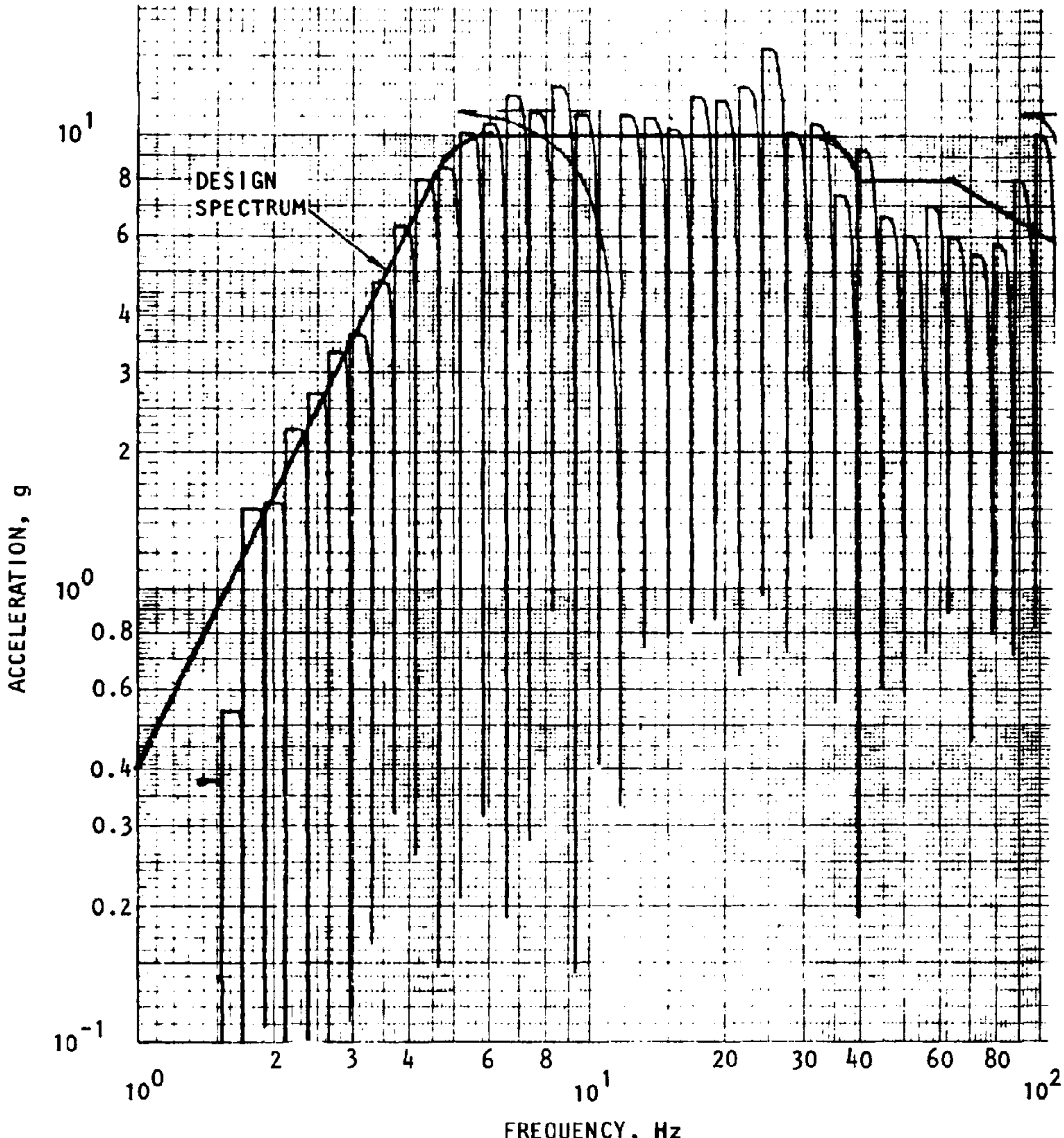

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FIGURE A-3A-1. COMPARISON OF DESIGN AND TEST RESPONSE SPECTRA, HORIZONTAL AXIS

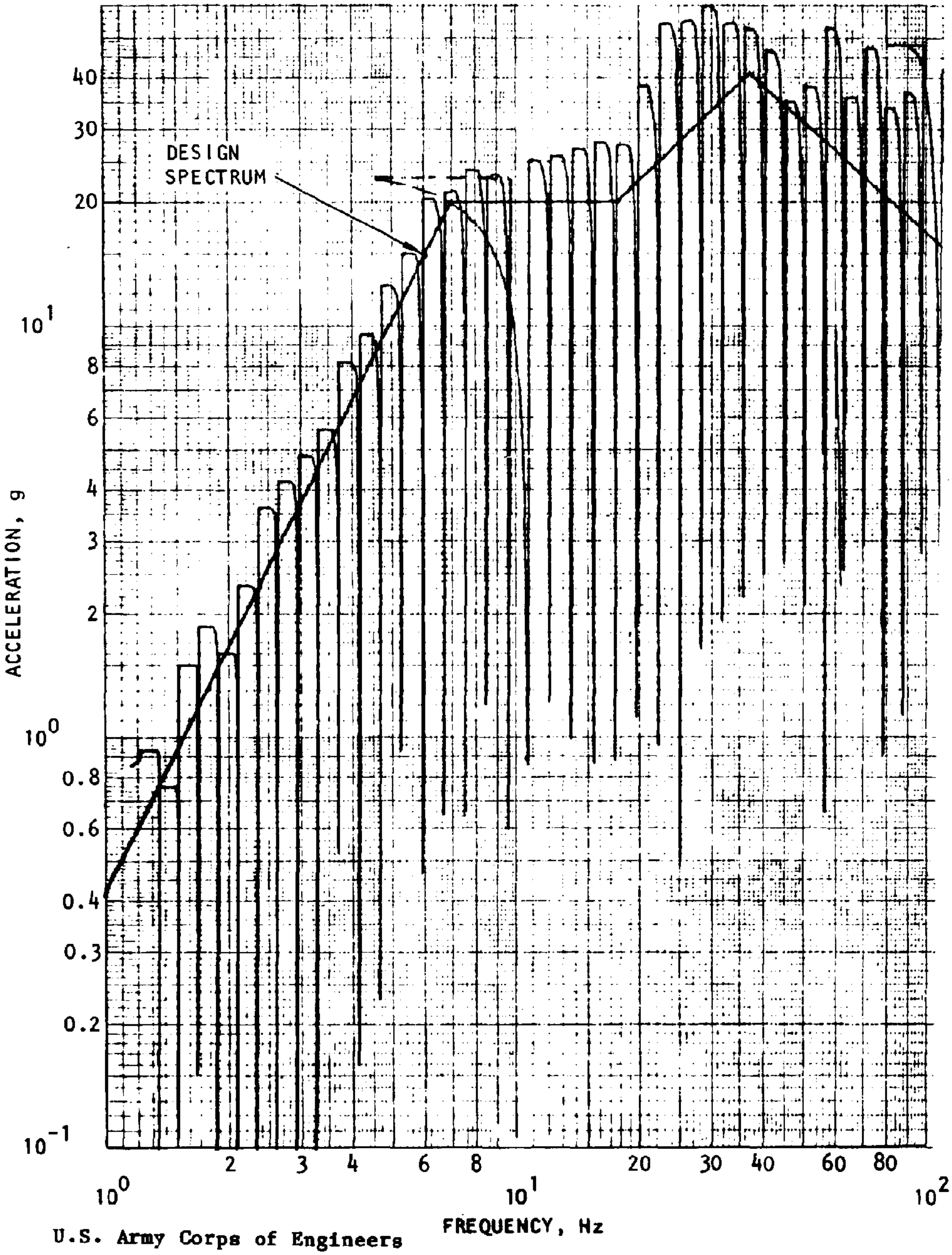

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FIGURE A-3A-2. COMPARISON OF DESIGN AND TEST RESPONSE SPECTRA, VERTICAL AXIS

A-3B. GENERATOR CONTROL PANEL

1. Generator control panel, (cabinet). General Electric Co., E01GD, 26 in. wide, 26 in. long, 89 in. high, 800 lb.

2. Test.

a. Shock spectra.

Horizontal	Figure A-3B-1
Vertical	Figure A-3B-2

b. Single axis tests.

3. Results. Passed with following exception: At the completion of final shock No. 17, the Agastat time delay relay was removed from the generator control panel and disassembled. An examination of the contacts, wipers, and associated circuitry was performed. This examination revealed that the spring tension of wiper which controls contact terminals 2, 4, and 6 was weaker than adjacent wiper which controls contact terminals 1, 3, and 5. The contact surfaces were inspected and cleaned. The time delay relay was then reassembled and a continuity check performed. This check revealed that all contacts were now operating in accordance with the test specifications; however, it was noted that any movement of the relay would result in a momentary interruption of contact which controls terminals 4 and 6.

4. References and credits.

Army Corps of Engineers (COE). *Test Report on Safeguard Shock Environmental Testing of Generator Control Panel*, HNDSP-74-310-ED-R. Huntsville, AL: COE, Jun 1974.

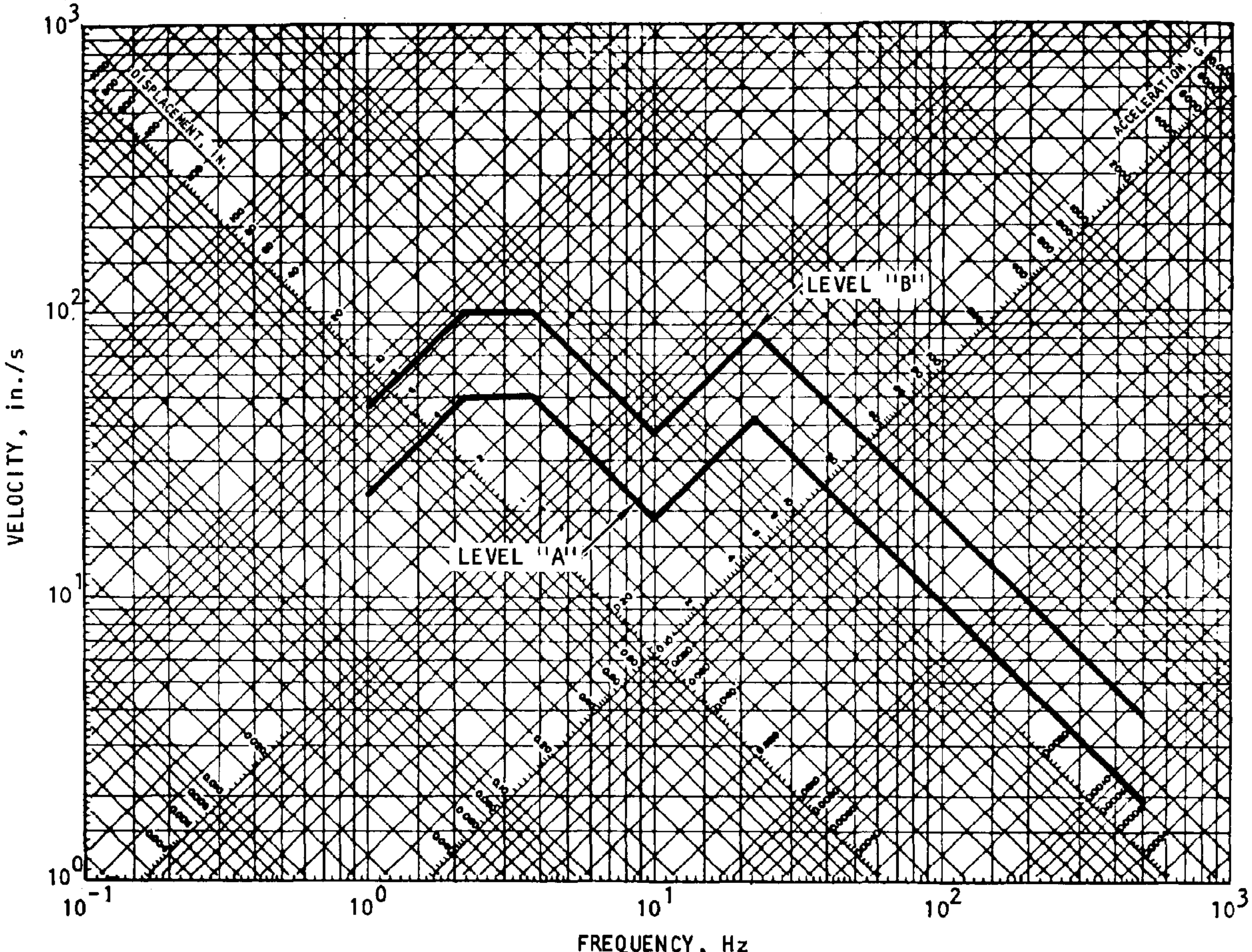

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FIGURE A-3B-1. TRANSIENT HORIZONTAL SHOCK SPECTRA TEST CRITERIA

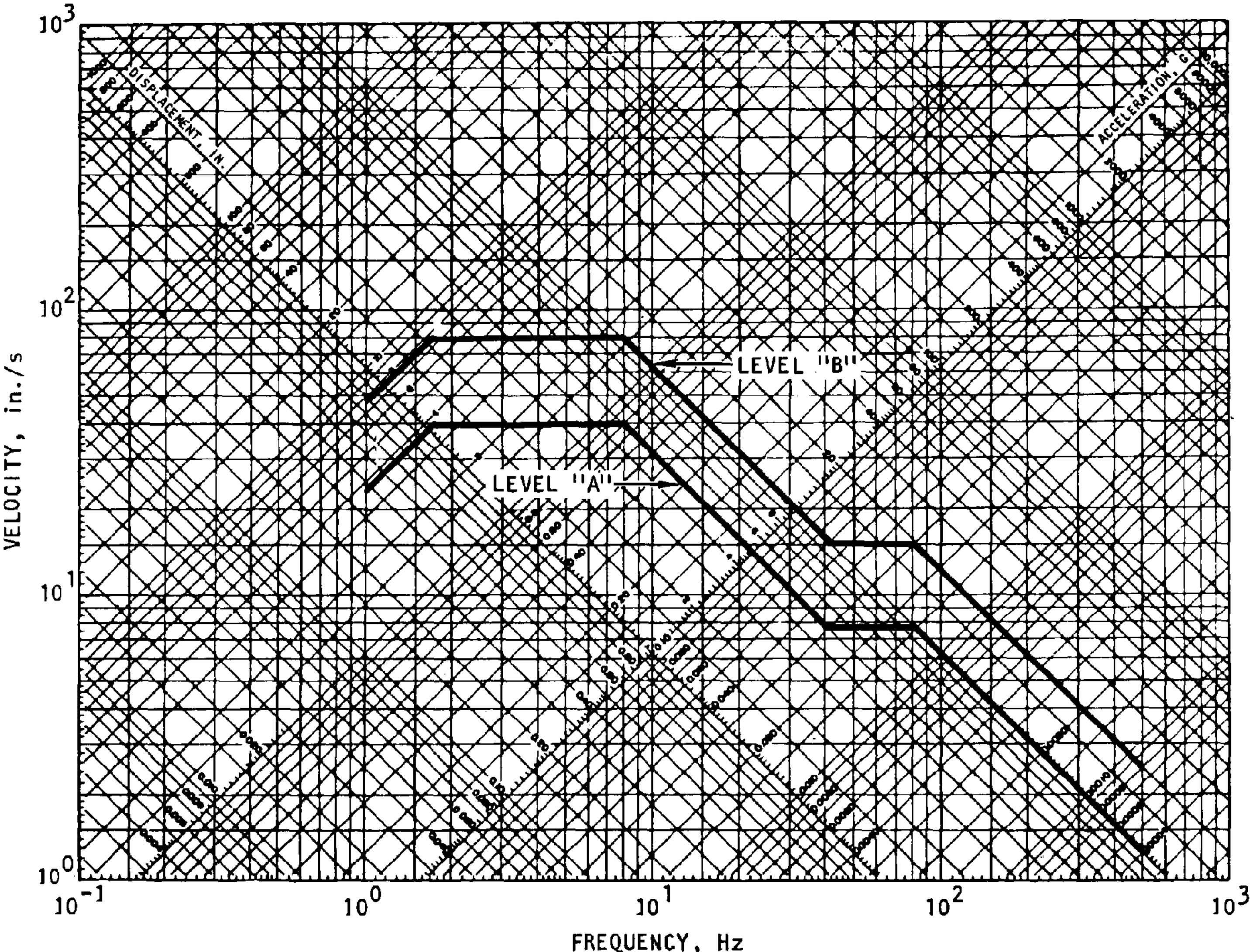

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FIGURE A-3B-2. TRANSIENT VERTICAL SHOCK SPECTRA TEST CRITERIA

A-3C. ELECTRIC MOTOR-GENERATOR CONTROL CENTER

1. Electric motor-generator control center, Item Type Code (ITC) E03GM. Manufactured by Bogue Electric Company. The five-cabinet unit is 12 ft wide, 4 ft 6 in. deep, and 7 ft 6 in. high. The separately mounted cabinet is 3 ft wide, 4 ft deep, and 8 ft 6 in. high. Equipment weight is approximately 9336 lb.

2. Test.

a. Shock spectra.

Horizontal	Figure A-3C-1
Vertical	Figure A-3C-2

b. Biaxial tests.

3. Results.

a. Problems experienced.

(1) Most of the functional failures that occurred during the testing involved the appearance of red flags on the relays, indicating that the relays had been tripped. This was not true, however, and it was determined that the flags were shaken loose during the testing.

(2) The only mechanical damage that occurred was the shearing of the bolts that latched the front doors of the generator cabinet and the distribution cabinet during test 12, 100 percent X-Z. All six bolts were replaced and testing continued.

(3) During test 21, 100 percent Y-Z, the bus tie breaker in the bottom of the bus tie cabinet changed state. Since the breaker operated satisfactorily during postshock testing, it was reset and testing continued.

(4) Throughout the tests, contacts of seal-in units and/or indicating contractor switches of protective relays chattered closed for various times.

b. Discussion of functional and structural anomalies.

(1) The shaft of the differential phase C relay was bent. This problem caused the differential phase C relay to be totally inoperative, making it impossible for the motor-generator set to detect a phase C ground fault or load imbalance; this could be fatal to the system. However, this fault did not occur during the testing.

(2) The bolts on the generator cabinet and distributor cabinet doors were sheared. This problem could cause substantial chatter on the relay contacts mounted on these doors. In addition, the doors could become completely detached from the test specimen, severing cables and therefore disabling fault indication and control.

(3) The main tie breaker had a change of state. Dropout of the main tie breaker, a critical system failure, was caused by chatter on one or several of the relays associated with its protection circuit. Appearance of flags was probably caused more by mechanical stress than actual contact closure, since flags appeared when there was no chatter on the associated contacts. Consequently, appearance of a flag does not necessarily indicate an electrical functional anomaly. Contact chatter has an adverse degrading effect on the normal functioning of protective relays. When chatter continues long enough, tripping of protected circuit breakers is initiated.

4. References and credits.

Army Corps of Engineers (COE). *Test Report on Safeguard Shock Environment Test of Motor-Generator Set E03GM*, HNDSP-74-337-ED-R. Huntsville, AL: COE, Dec 1974.

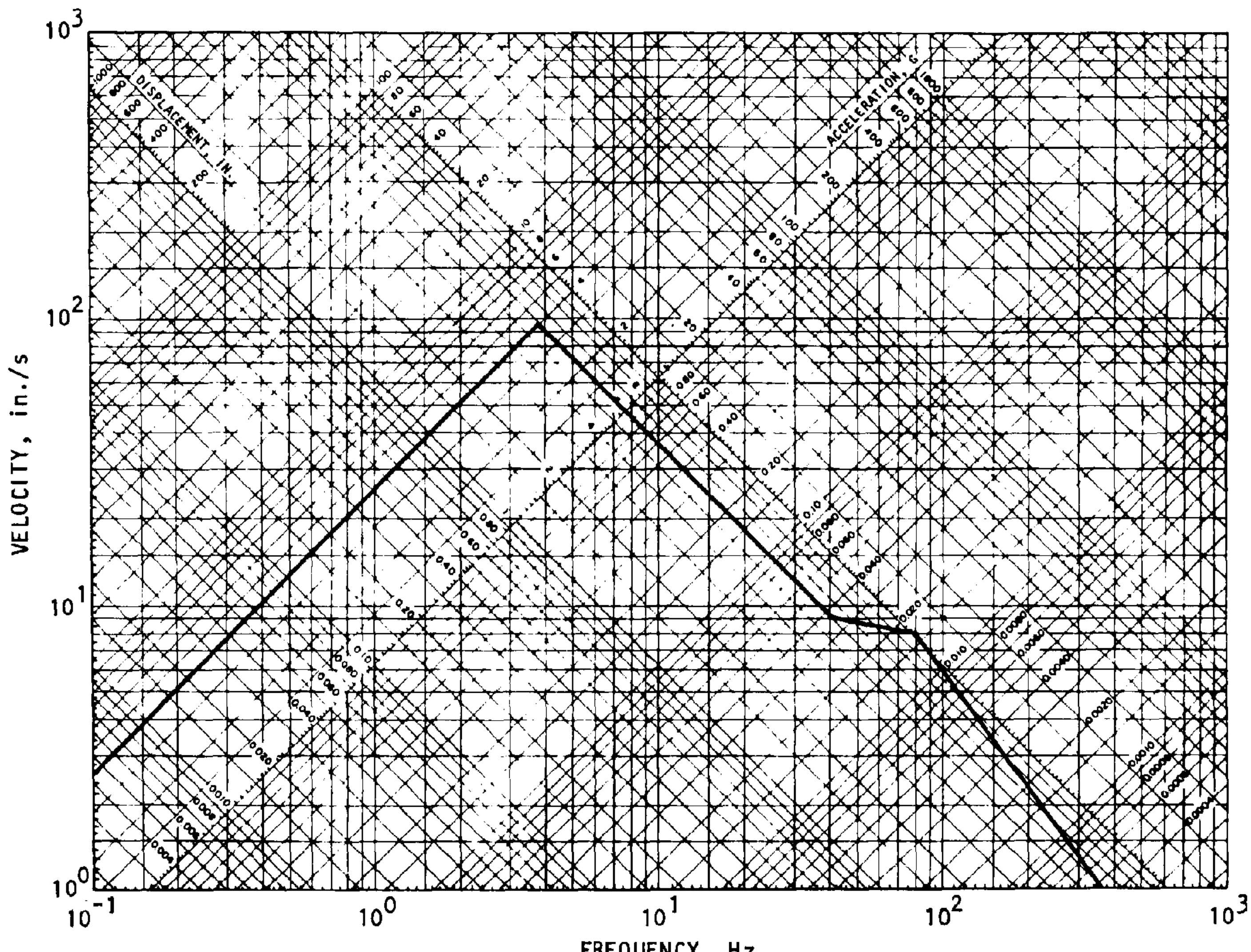

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FIGURE A-3C-1. HORIZONTAL SHOCK SPECTRUM

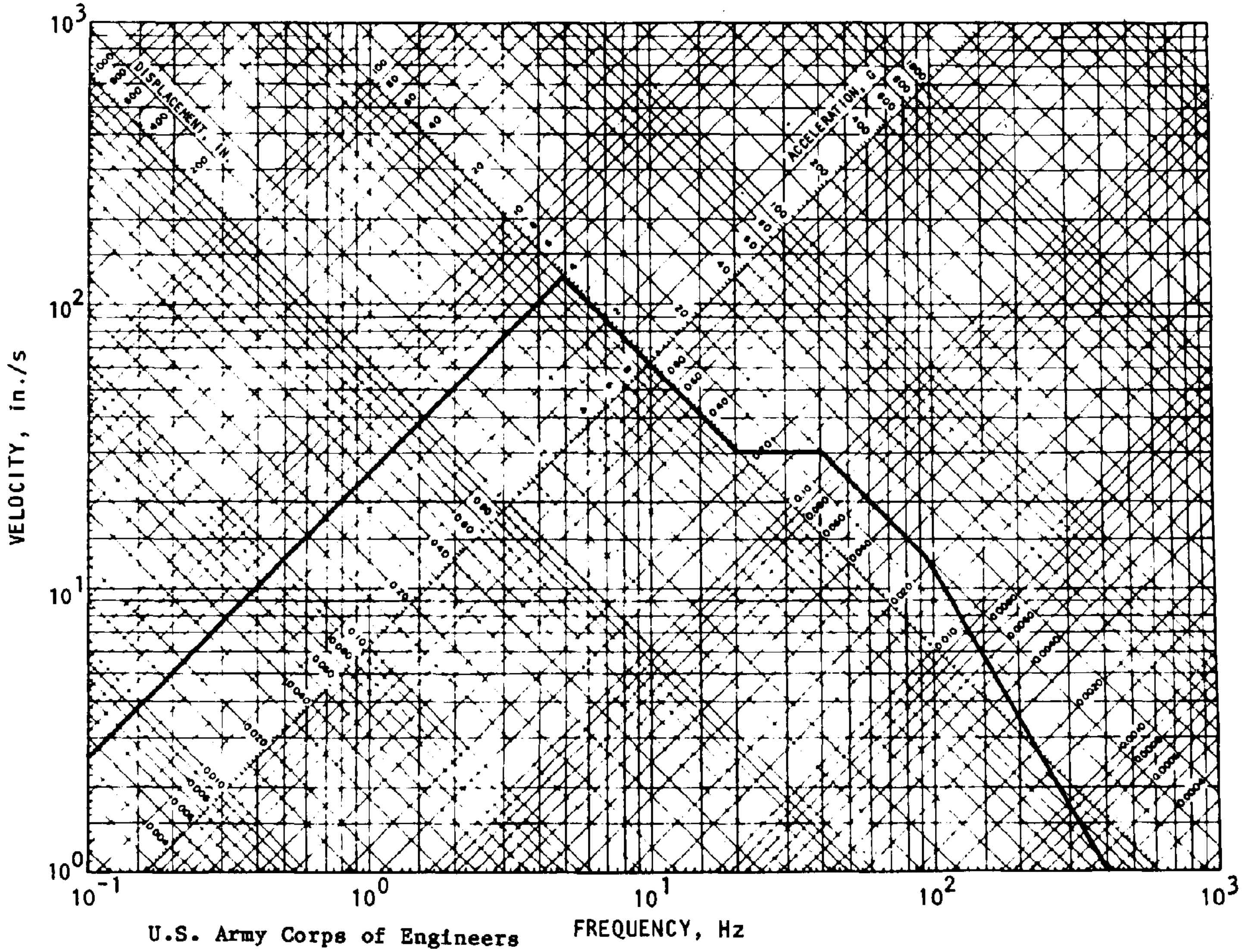

FIGURE A-3C-2. VERTICAL SHOCK SPECTRUM

A-3D. ELECTRIC MOTOR CONTROL CENTER

1. Electric motor control center, Item Type Code (ITC) E12SS. A pair of cabinets manufactured by the Westinghouse Electric Corporation, bolted together to form a unit. The equipment is 3 ft 6 in. wide, 5 ft 6 in. deep, and 7 ft 10 in. high. Control center weighs 722 lb. One cabinet contains four starters and the other six.

2. Test.

a. Shock spectra.

Vertical and horizontal Figure A-3D-1

b. Axial tests.

3. Results.

a. Deviations from proper performance. Instances where the breaker contacts lost continuity during the shock motions were observed. All breakers functioned normally at the completion of the testing.

b. Discussion of functional anomalies. Chatter in the contacts of the starters would not seriously affect operation of the equipment controlled by the breakers. The longest duration of chatter, 110 msec, is so short that the system function accomplished through the connected load would not have enough time to deviate appreciably. Arcing between the contacts on separation might cause erosion of the surfaces, but the tendency to weld the contacts together would be minimal or nonexistent.

4. References and credits.

Army Corps of Engineers (COE). *Test Report on Safeguard Shock Environment Test of Motor Control Center E12SS*, HNDSP-74-333-ED-R. Huntsville, AL: COE, Dec 1974.

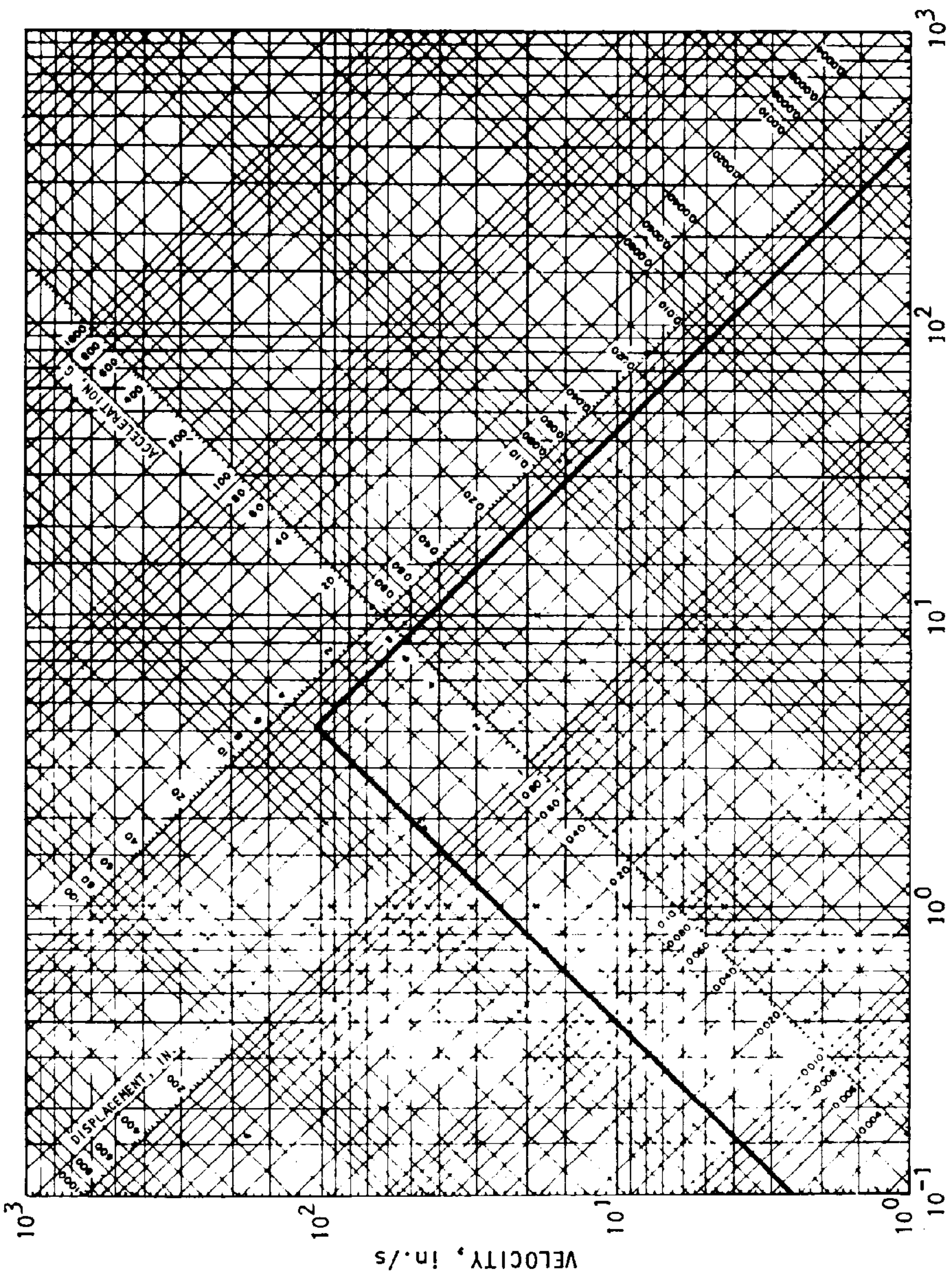

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FIGURE A-3D-1. VERTICAL AND HORIZONTAL SHOCK SPECTRUM

A-3E. AIR COMPRESSOR, CONTROL PANEL, AND DRIVE MOTOR

1. Air compressor, control panel, and drive motor. Control cabinet (Chicago Pneumatic) weighs 422 lb and measures 12 in. by 35 in. by 63 in.; drive motor (U.S. Electric Motor) weighs 1105 lb and measures 33 in. by 30 in. by 25 in.; and temperature switch (United Electronics) weighs 5 lb and measures 8 in. by 5 in. by 2 in.

2. Test.

a. Shock spectra.

Horizontal	Figure A-3E-1
Vertical	Figure A-3E-2

b. Single axis tests.

3. Results.

a. During shock No. 11, 75 percent level, vertical axis, the temperature switch contacts momentarily opened causing the hi-after cooler air temperature fault to trip. This electrical trip caused the system to shut down. It was also noted that the disable vibration switch tripped causing a change from normal to a disable mode as indicated by a red warning light on the face of the control panel.

b. During shock No. 14, 100 percent level, vertical axis, the disabled vibration switch tripped causing a change from a normal to a disable mode as indicated by a red warning light on the face of the control panel. This anomaly did not affect the operation of the system. There was no other functional degradation noted.

4. References and credits.

Army Corps of Engineers (COE). *Test Report on Safeguard Shock Environmental Testing of Air Compressor Control Panel, Drive Motor*, HNDSP-74-309-ED-R. Huntsville, AL: COE, May 1974.

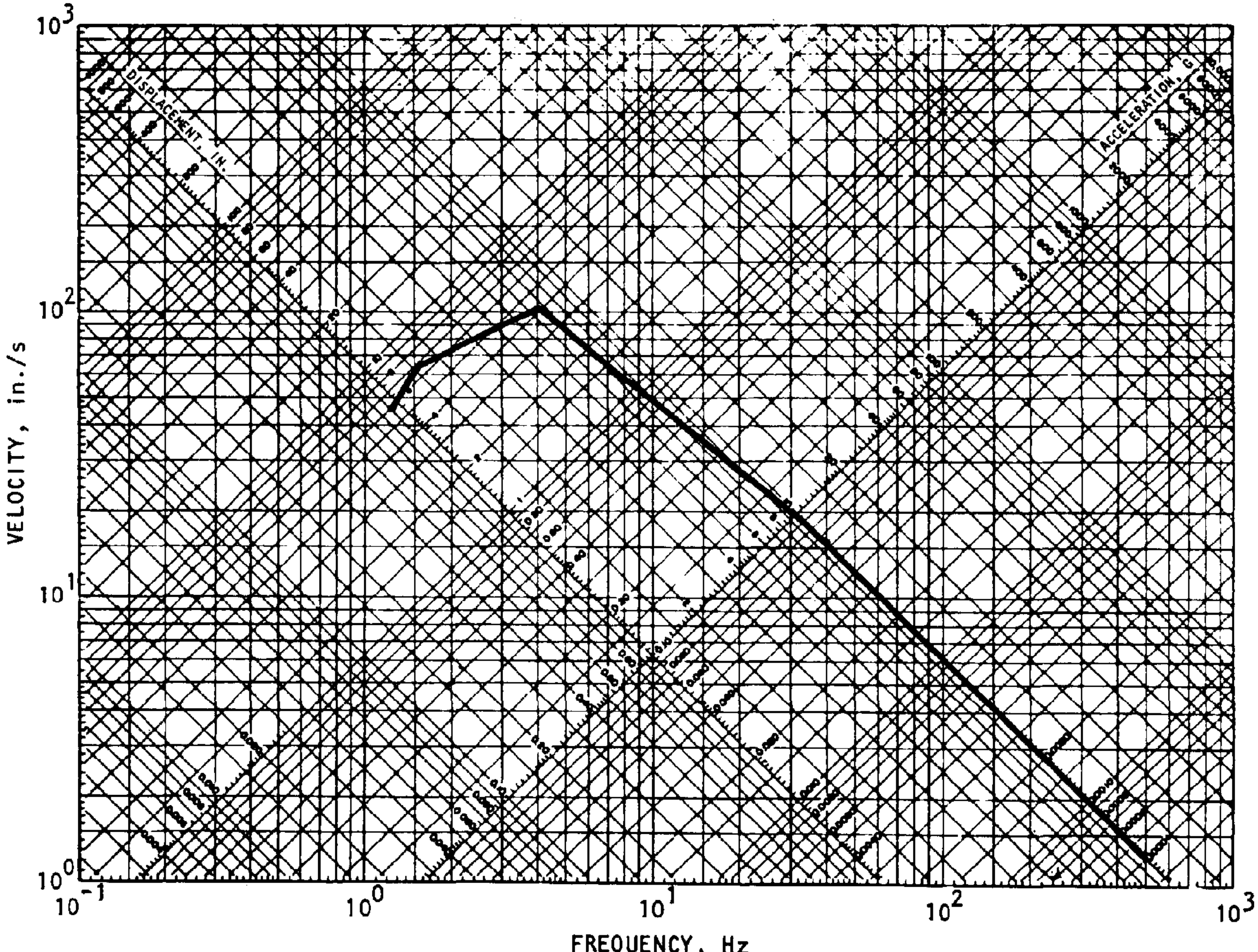

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FIGURE A-3E-1. HORIZONTAL SHOCK SPECTRA

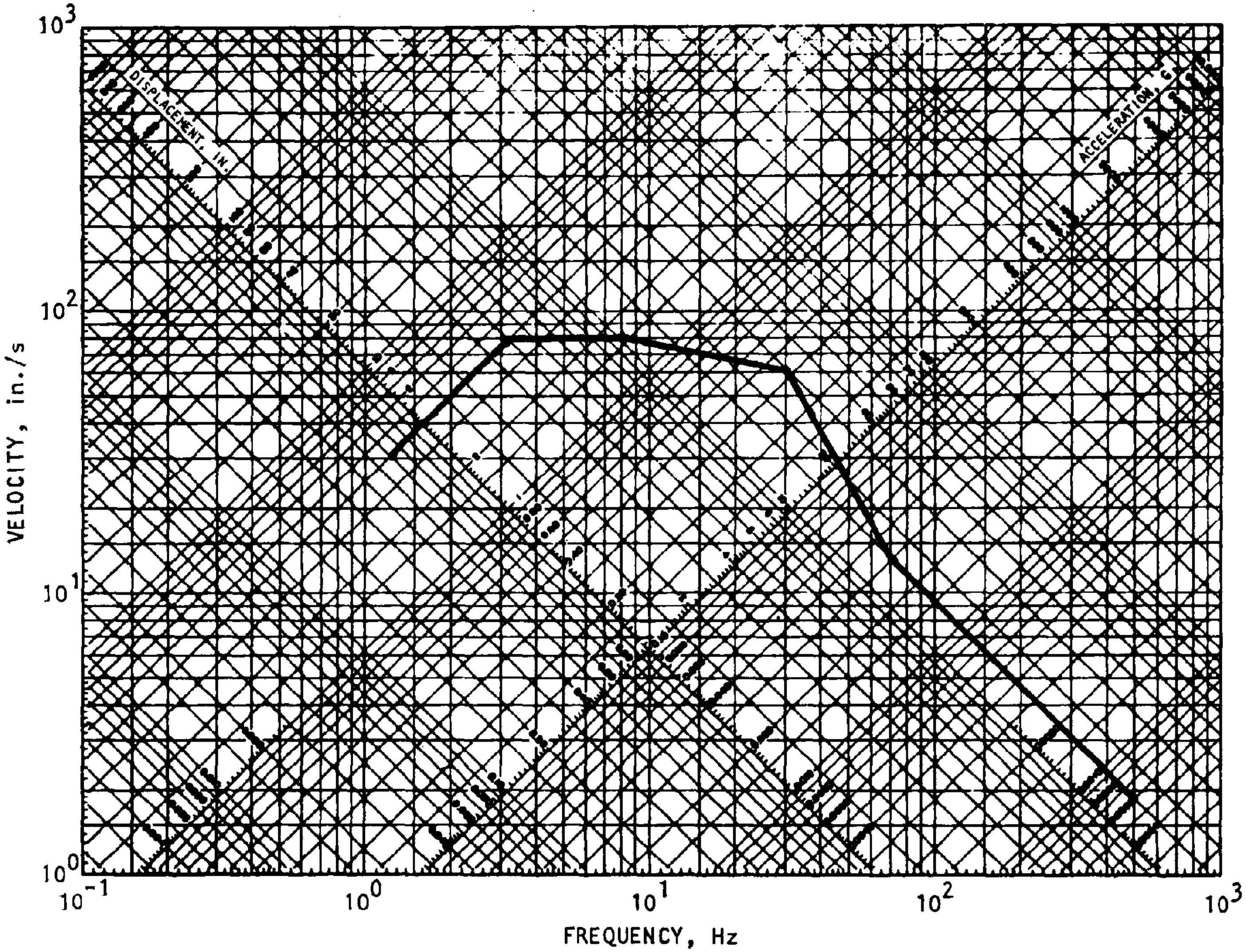

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FIGURE A-3E-2. VERTICAL SHOCK SPECTRA

A-3F. MOTOR CONTROL CENTER

1. Motor control center. An assembly of five commercial grade standard motor control cabinets, made by Westinghouse Electric Corporation. The assembly measures 7 ft 11 in. long, 20 in. wide, and 7 ft 6 in. high. Total cabinet weight is 1870 lb.

2. Test.

a. Shock spectra.

Horizontal and vertical — Figure A-3F-1

b. Axial tests.

3. Results. No mechanical or functional failures occurred during the shock testing of test specimen. Inspection of continuity sensor data revealed no instances of contact change of state.

4. References and credits.

Army Corps of Engineers (COE). *Test Report on Safeguard Shock Environment Test of Electrical Motor Control Center ITC E05SS Specimen A*, HNDSP-74-331-ED-R. Huntsville, AL: COE, Dec 1974.

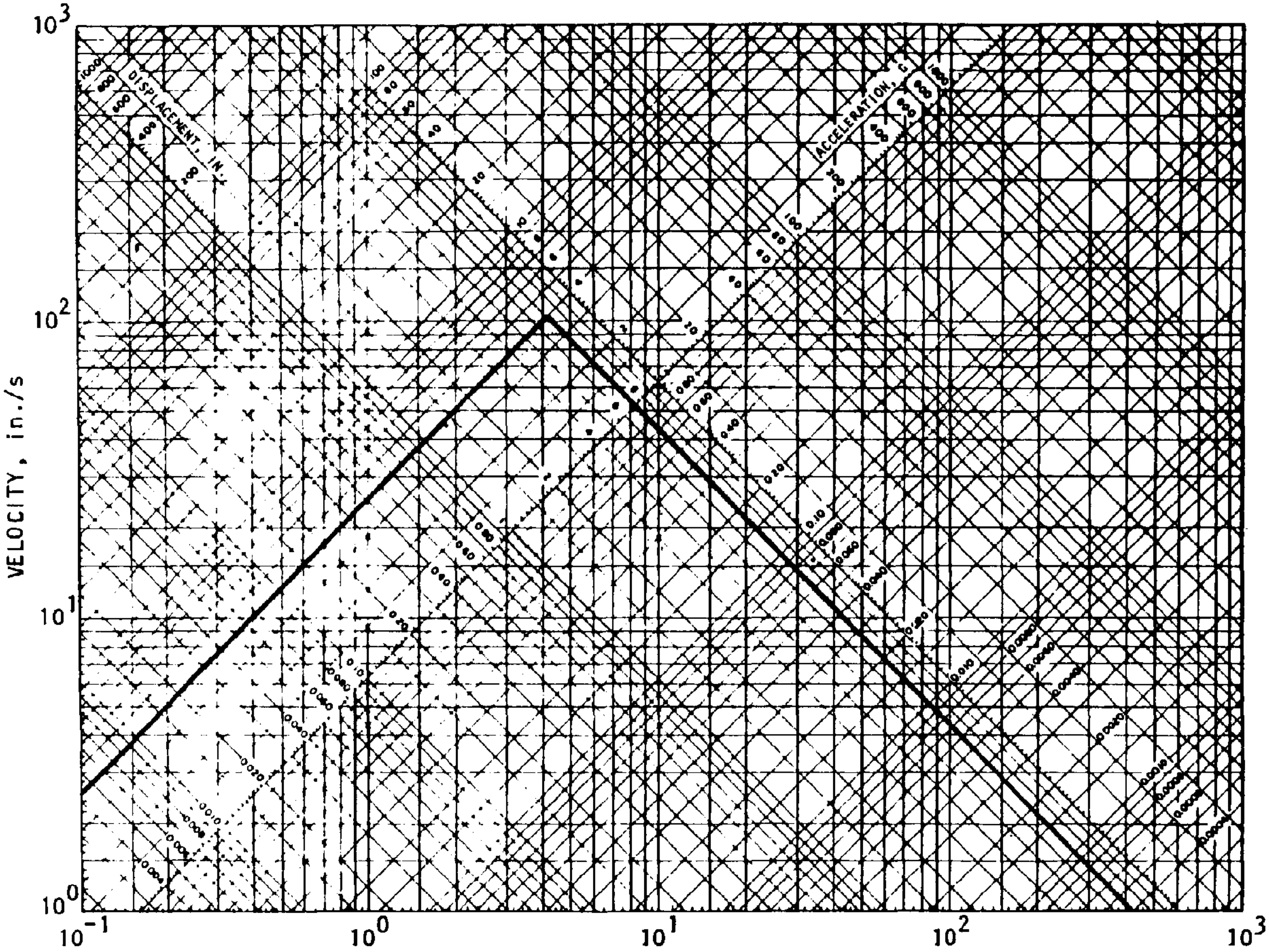

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FIGURE A-3F-1. VERTICAL AND HORIZONTAL SHOCK SPECTRUM

A-3G. ELECTRIC MOTOR CONTROL CENTER

1. Electric motor control center cabinets. Cabinet No. 1, 2231 lb, 90 in. high, 60 in. wide, 40 in. deep and Cabinet No. 2, 2146 lb, 102 in. high, 80 in. wide, 35 in. deep are manufactured by Hatch Manufacturing Co., El Paso, Texas.

2. Test.

a. Shock spectra.

Horizontal and vertical — Figure A-3G-1

b. Single axis tests.

3. Results. Table A-3G-1.

4. References and credits.

Army Corps of Engineers (COE). *Test Report on Safeguard Shock Environmental Testing of Electric Motor Control Centers (E52MC and E87MC)*, HNDSP-74-315-ED-R. Huntsville, AL: COE, Jul 1974.

Table A-3G-1. Test results

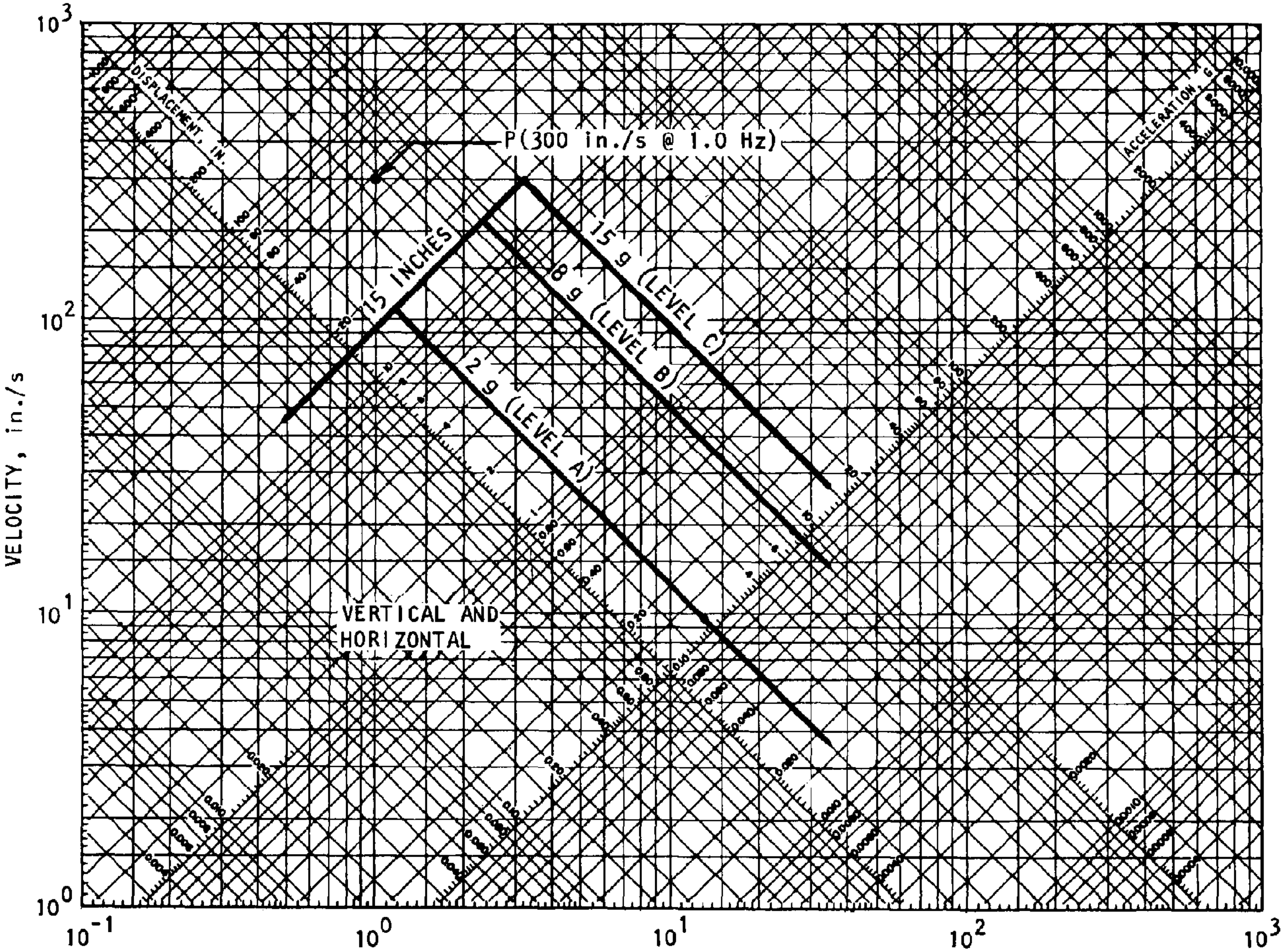
Specimen	Axis	Run No.	Test Level	Electrical Malfunction	Structural Damage	Remarks
E87MC	Z	1-1	A	No	No	
E87MC	Z	1-2	A	No	No	
E87MC	Z	1-3	A	No	No	
E87MC	Z	2-1	B	No	No	
E87MC	Z	2-2	B	No	No	
E87MC	Z	3-1	C	Yes	No	Chilled water pump starter tripped. Jumper installed on relay R3 between Terminals 5 and 10.
E87MC	Z	3-2	C	Yes	No	Air conditioner overload relay tripped--manually reset.
E87MC	Z	4	4" DA	No	No	
E87MC	X	5	4.3" DA	No	No	
E87MC	X	6-1	A	Yes	No	Chilled water pump breaker tripped--manually reset.
E87MC	X	6-2	A	No	No	
E87MC	X	6-3	A	No	No	
E87MC	X	6-4	A	Yes	No	Chilled water pump breaker tripped--manually reset.
E87MC	X	7-1	B	Yes	No	Jumper installed on chilled water pump relay R3 between Terminals 5 and 10. Lost "run" condition. Indicated "ready."
E87MC	X	7-2	B	No	No	
E87MC	X	8-1	C	Yes	No	ACU change "run" to "ready." Jumper installed on ACU relay R3 Terminals 5 and 10
E87MC	X	8-1A	-	No	No	75% level C (rerun)
E87MC	X	8-2	C	No	Yes	Welds cracked on mounting channel at vertical member joints.
E87MC	Y	9-*	42" DA	No	No	
E87MC	Y	10-1	A	No	No	
E87MC	Y	10-2	A	No	No	
E87MC	Y	10-3	A	Yes	No	ACU and chilled water pump circuits changed from "run" to "ready."
E87MC	Y	11-1	B	No	No	50% Level B

Table A-3G-1. Test results (continued)

Specimen	Axis	Run No.	Test Level	Electrical Malfunction	Structural Damage	Remarks
E87MC	Y	11-2	B	Yes	No	ACU and chilled water pump circuits changed from "run" to "ready." Jumpers installed on relay R3 between Terminals 5 and 10.
E87MC	Y	12-1	-	No	No	75% Level C
E87MC	Y	12-2	C	No	No	
E87MC	Y	12-3	-	No	Yes	125% Level C. Welds on mounting channel cracked. Rivets failed on relay shelves.
E52MC	Z	13	4.0" DA	No	No	
E52MC	Z	14-1	-	No	No	50% Level A
E52MC	Z	14-2*	A	No	No	
E52MC	Z	14-3	A	No	No	
E52MC	Z	15-1	B	Yes	No	Contact chatter on fire water pumps.
E52MC	Z	15-2	B	Yes	No	Chatter on fire water pump 7501. Manual lock on 7502.
E52MC	Z	15-3	B	Yes	No	Manual lock removed. Chatter on 7502. Removed motor starter coil connection on 7501.
E52MC	Z	16-1	C	Yes	No	Chatter on pump 7502. Reinstalled coil connection on 7501. Removed manual lock.
E52MC	Z	16-2	C	Yes	No	Lost screw connecting L2 to control transformer P7501 changed state.
E52MC	Z	16-3	C	Yes	No	Chatter on pump 7502. Engaged manual lock.
E52MC	Z	16-4	C	No	No	
E52MC	X	17	42."DA	No	No	
E52MC	X	18-1	-	No	No	50% Level A
E52MC	X	18-2	A	No	No	
E52MC	X	18-3	A	No	No	
E52MC	X	19-1	B	Yes	No	Chatter on pump 7502. Engaged manual lock.
E52MC	X	19-2	B	No	No	Remove manual lock.

Table A-3G-1. Test results (concluded)

Specimen	Axis	Run No.	Test Level	Electrical Malfunction	Structural Damage	Remarks
E52MC	X	20-1	C	Yes	No	Lost air compressor output, hot water pump, domestic water pump, fire pump 7502.
E52MC	X	20-2	C	Yes	No	Chatter on fuel oil unloading area, test tower, ind. sump pump, eject steam boiler, and fire water pump 7502.
E52MC	Y	21	4.0" DA	No	No	
E52MC	Y	22-1	-	Yes	No	50% Level A. Chatter on pump 7502. Engaged manual lock.
E52MC	Y	22-2	A	No	No	No test (restart computer).
E52MC	Y	22-1	-	No	No	50% Level A. Remove manual lock.
E52MC	Y	22-2	A	No	No	
E52MC	Y	22-3	A	Yes	No	Chatter on pump 7502.
E52MC	Y	23-1	B	Yes	No	Chatter on pump 7502. Engage lock.
E52MC	Y	23-2	B	No	No	Release lock.
E52MC	Y	24-1	C	Yes	No	Chatter on 7501 and 7502. Lost air compressor output oil valve tripped. Engaged lock on pump 7502.
E52MC	Y	24-2	C	Yes	No	Chatter on 7501. Lost air compressor output starter oil relay tripped.



U.S. Army Corps of Engineers FREQUENCY, Hz

FIGURE A-3G-1. SHOCK SPECTRA

A-3H. CIRCUIT BREAKERS

1. Unit substation, circuit breakers. The electrical equipment was manufactured by the Westinghouse Electric Corporation for use on three-phase, 60-Hz alternating current. The substation is 5 ft 3 in. wide, 5 ft 6 in. deep, and 7 ft 10 in. high. Equipment weight is 4949 lb.

2. Test.

a. Shock spectra.

Horizontal and vertical	Figure A-3H-1

b. Biaxial tests.

3. Results.

a. Problems experienced. No structural failures occurred during the testing, but there was interference with normal functioning of the equipment composing the substation. Loss of the meter cover during test 19 did not interfere with operation of the meter.

b. Deviations from proper performance. There were instances where the breakers failed to function normally.

c. Discussion of functional anomalies. Two types of anomalies occurred during the tests. The holder of the disconnecting contacts mounted on a circuit breaker broke, and some of the spring-loaded contacts in the holder were dislodged. The electrical control circuit to the tie breaker was lost, rendering the breaker unable to perform its switching function. The other anomaly was caused by rely contact chatter tripping circuit breakers and disconnecting loads.

4. References and credits.

Army Corps of Engineers (COE). *Test Report on Safeguard Shock Environment Test of Circuit Breakers E27SS*, HNDSP-74-335-ED-R. Huntsville, AL: COE, Dec 1974.

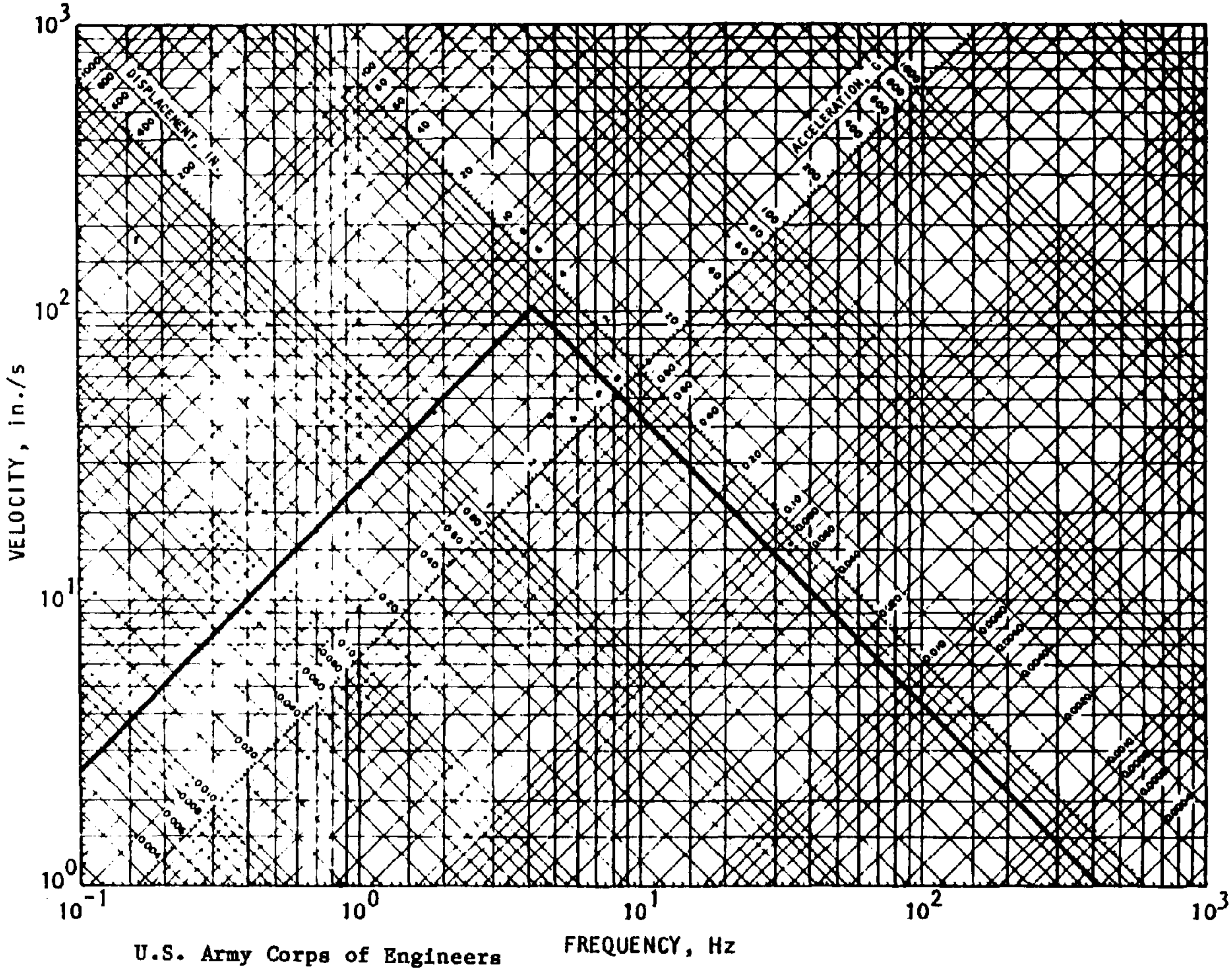


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FIGURE A-3H-1. VERTICAL AND HORIZONTAL SHOCK SPECTRUM

A-31. CIRCUIT BREAKER ASSEMBLY

1. Circuit breaker cabinet assembly. A standard commercial assembly made by Westinghouse Electric Corporation. The assembly is 8 ft 9 in. wide, 5 ft 6 in. deep, and 7 ft 10 in. high.

2. Test.

a. Shock spectra.

Horizontal and vertical   Figure A-3I-1

b. Biaxial tests.

c. Test machine: Electrohydraulic shaker (CERL facility)

3. Results.

a. Problems experienced.

(1) Damage during the shock testing was confined to the breakers. During run 2, 100 percent Y-A, the contacts between breaker unit 5, cubicle A, and the cabinet were bent so they could not make contact. The breaker would not function automatically, but did work manually.

(2) During the next run, 3, 100 percent Y-Z, breaker unit 3, cubicle C, tripped when a screw holding the trip shaft end plate came out causing the trip shaft to become dislodged. After the end plate and screw were replaced, the breaker functioned properly. During the same test, breaker unit 5, cubicle D, tripped due to internal mechanical failure. The breaker would not function manually or automatically. No other damage occurred during the Y-Z axes testing.

(3) After test 11, the contacts on breaker 5, cubicle A, were replaced with the contacts from the broken breaker in unit 5, cubicle D. After the next test, 50 percent X-Z, breaker unit 5, cubicle A, would not function automatially, but would function manually. Inspection revealed that the breaker disconnecting contact 1 had cracked so contact 1 could not make contact. During test 15, 75 percent X-Z, the contacts of breaker unit 5, cubicle A, were bent, and the plastic supporting them cracked. The breaker would not function automatically, although it did function manually.

b. Deviations from proper performance.

(1) Instances where the circuit breakers failed to perform normally are noted. The breaker change of state occurred in test 17, 100 percent level.

(2) In addition, the cabinets' structural fasteners started to loosen near the end of the test program.

c. Discussion of functional and structural anomalies.

(1) Failure of the circuit breakers to operate would have one of two results:

(a) If the breaker opened upon failure, all essential loads would be disconnected, impacting normal functioning of the system.

(b) If the breaker remained closed and inoperable, normal system operation would not be impacted unless an electrical fault developed on the related connected system, and the breaker would be unable to trip.

(2) Loosening of the cabinet door fasteners on this specimen reflected no appreciable structural degraduation.

4. References and credits.

Army Corps of Engineers (COE). *Test Report on Safeguard Shock Environment Test of Circuit Breakers E29SS*, HNDSP-74-336-ED-R. Huntsville, AL: COE, Dec 1974.

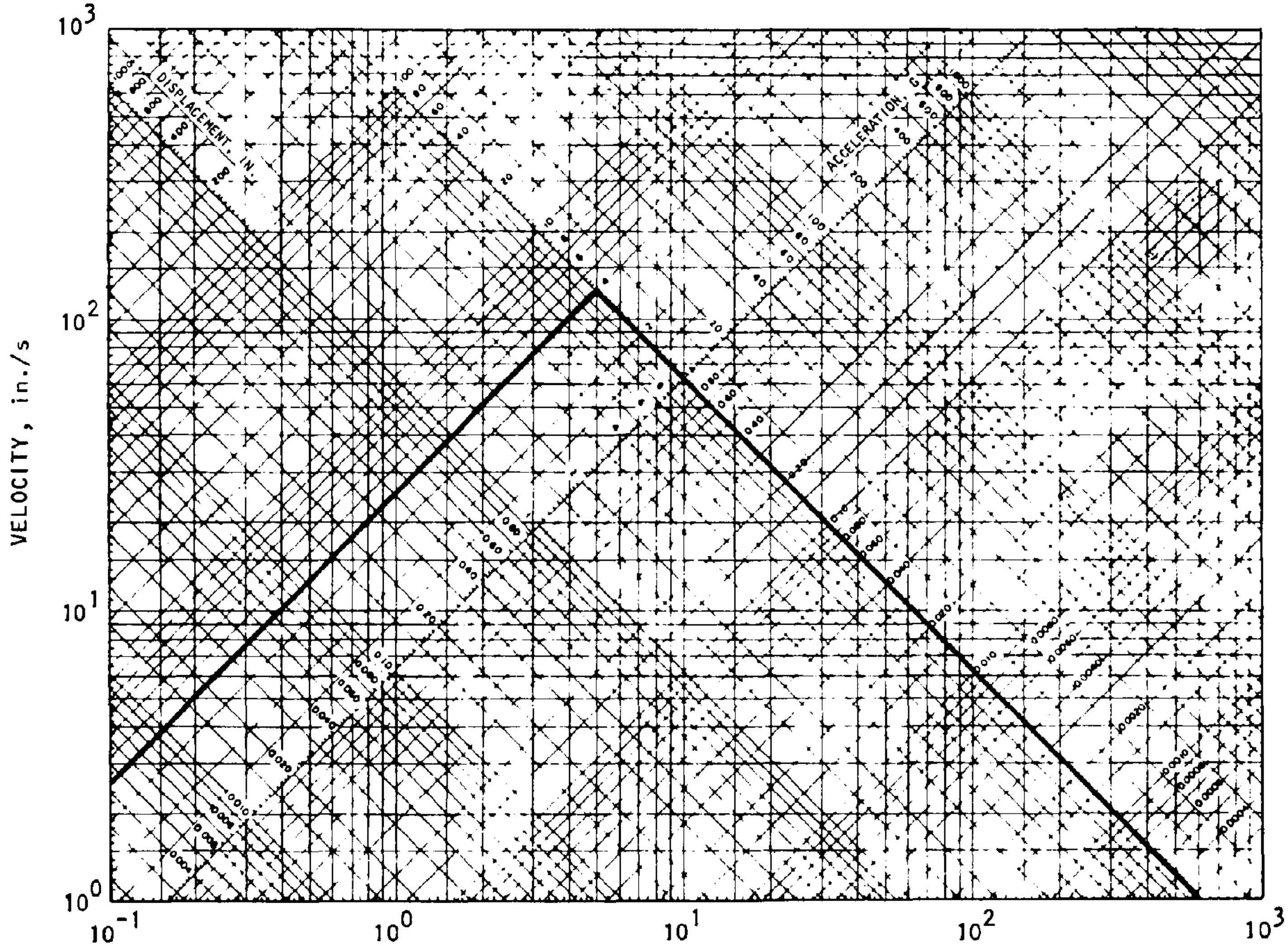

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FIGURE A-3I-1. VERTICAL AND HORIZONTAL SHOCK SPECTRUM

A-3J. VOLTAGE REGULATOR AND CONTROL CENTER

1. Voltage regulator and motor control center assembly. A Westinghouse Electric Corporation standard unit. It is 6 ft 9 in. long, 5 ft 6 in. deep, and 7 ft 10 in. high. Substation weighs 4771 lb and consists of three cabinets, one each for circuit breakers, meters, and voltage regulator.

2. Test.

a. Shock spectra.

Horizontal and vertical — Figure A-3J-1

b. Biaxial tests.

3. Results.

a. Problems experienced. No structural failures occurred during the testing, but interference with normal functioning of two of the components did occur. Device 97-2 chattered during runs 5, 6, and 8, at levels of 75, 75, and 100 percent respectively. When checked after these tests, the breaker performed normally. The flag on relay 27-2X dropped during runs 19, 20, and 21, all 100 percent level. When checked after these tests, the relay performed normally after the flag was reset.

b. Discussion of functional anomalies. The flag on relay 27-2X that dropped is an indicating device; its malfunction had no adverse effect on the circuit design function of the relay. After the breaker tripped several times due to abnormal functioning of device 97-2, investigation revealed that associated wiring was tangled about the breaker so that its functioning became unstable. After correction of the wiring problem, testing continued and the anomaly did not recur.

4. References and credits.

Army Corps of Engineers (COE). *Test Report on Safeguard Shock Environment Test of Components of Unit Substation E05SS—Specimen B (Voltage Regulator and Circuit Breaker Section)*, HNDSP-74-332-ED-R. Huntsville, AL: COE, Dec 1974.

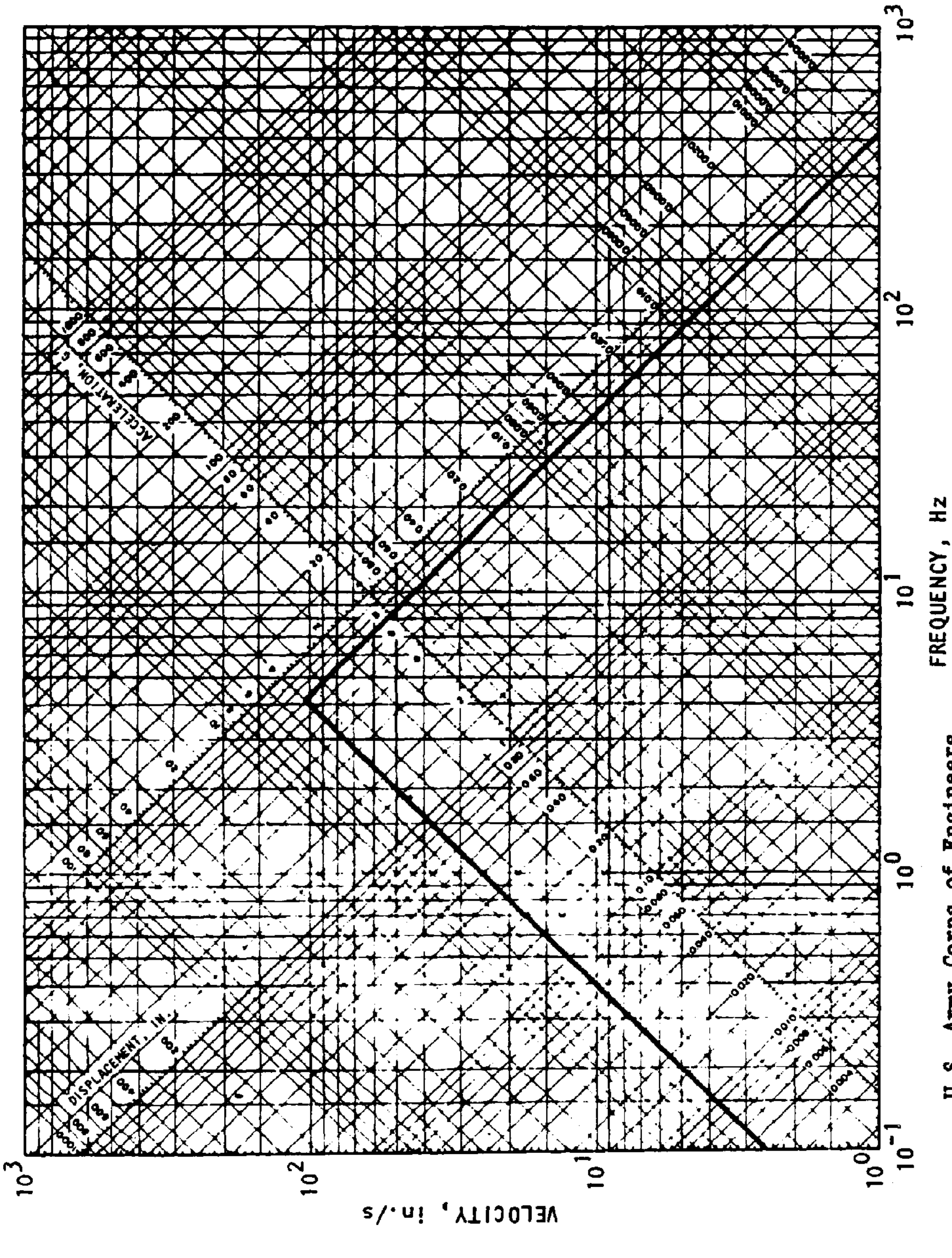

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FIGURE A-3J-1. VERTICAL AND HORIZONTAL SHOCK SPECTRUM

A-3K. UNIT SUBSTATION (SWITCH/TRANSFORMER)

1. Unit substation.

a. A Westinghouse Electric Corporation Model SL, serial number (S/N) PBV 2241-02. The switch is rated 600 amps at 5000 V. The transformer is rated 168 kVA at 65°C rise, 4160 to 480 Y V. The equipment is designed for use on three-phase 60 Hz alternating current. The substation is 11 ft long, 6 ft long, 1½ in. wide, 8 ft 4½ in. high, and weighs 8800 lb.

b. The equipment consists of a high-voltage switch section, a transformer section, and a dual section consisting of an output voltage regulator and output terminal cubicles.

2. Test.

a. Shock spectra.

Horizontal and vertical	Figure A-3K-1

b. Biaxial test.

3. Results.

a. Problems experienced. No structural failures occurred during the testing, but interference with normal functioning of the equipment composing the substation did occur. The most serious functional failure was mechanical breaking of a fuse in the sensing circuit of the voltage regulator during test 14. This failure interrupted operation of the regulator, but there was no deviation in output voltage of the regulator being monitored. The condition was corrected by installation of a new fuse. The chatter observed in the high voltage switch during tests 4 and 6 did not interfere with the conduct of the test program, but its effect on the connected loads should be evaluated.

b. Discussion of functional anomalies.

(1) The effect of chatter in the high voltage switch on the actual operation of the unit substation would be a matter of extreme concern as to this type of circuit interruption on the connected loads. Depending on the construction of the switch, replacing the entire switch assembly may be necessary rather than attempting to repair or replace only the blades.

(2) The broken fuse in the regulator sensing circuit would cause the regulator to stop functioning.

4. References and credits.

Army Corps of Engineers (COE). *Test Report on Safeguard Shock Environment Test of Electrical Unit Substation E04SS*, HNDSP-74-330-ED-R. Huntsville, AL: COE, Dec 1974.

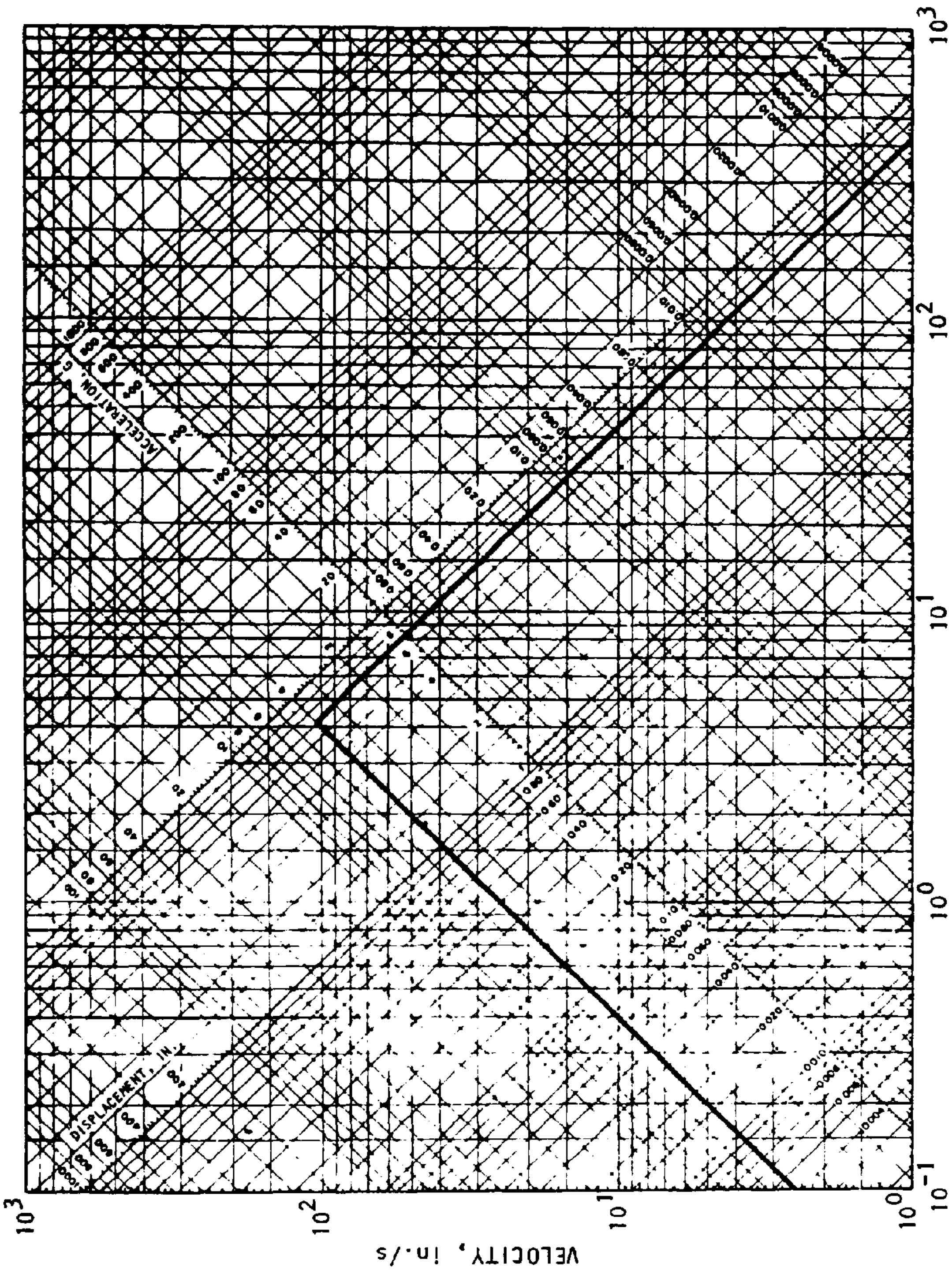

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FIGURE A-3K-1. VERTICAL AND HORIZONTAL SHOCK SPECTRUM

A-3L. UNIT SUBSTATION TRANSFORMER

1. Unit substation transformer. A Westinghouse Manufacturing Company Model SL, serial number (S/N) PBV 2233-06, rated 13,800 to 480 Y volts at 1000 kVA. The oil-filled transformer weighs 11,116 lb, and is 7 ft 9 in. wide, 6 ft 6½ in. deep, and 8 ft 10 in. high.

2. Test.

a. Shock spectra.

Horizontal	Figure A-3L-1
Vertical	Figure A-3L-2

b. Biaxial test.

3. Results.

a. Problems experienced.

(1) Consistent failure of the sudden pressure relay did not interrupt the progress of the testing. Resetting the relay after each test was the only repair needed before testing continued.

(2) The structural failure of the mounting brackets at the base of the transformer during test 20, a 100 percent X-Z axis test, caused a short delay in the test program. Repairs to the base necessitated removal of the transformer from the shake table.

b. Discussion of functional and structural anomalies.

(1) Dropping out of the sudden pressure relay under normal operating conditions would drop the load from the transformer, thus cutting off all power to everything dependent on it for power.

(2) Structural damage to the mounting brackets of the transformer could have extremely serious consequences, particularly if the transformer were thrown down. The electric shock hazard from broken cables would be serious until the electric supply could be cut off. The coolant within the transformer's fins is toxic, so rupture of the cooling system would also create a hazard to personnel.

(3) If the insulator were to break loose completely, fracture of the porcelain insulators could cause grounding of the primary leads to the transformer frame. The resulting heat and flame could lead to subsequent rupture of the cooling system, resulting in escape of the toxic coolant.

4. References and credits.

Army Corps of Engineers (COE). *Test Report on Safeguard Shock Environment Test of Unit Substation Transformer E16SS*, HNDSP-74-334-ED-R. Huntsville, AL: COE, Dec 1974.

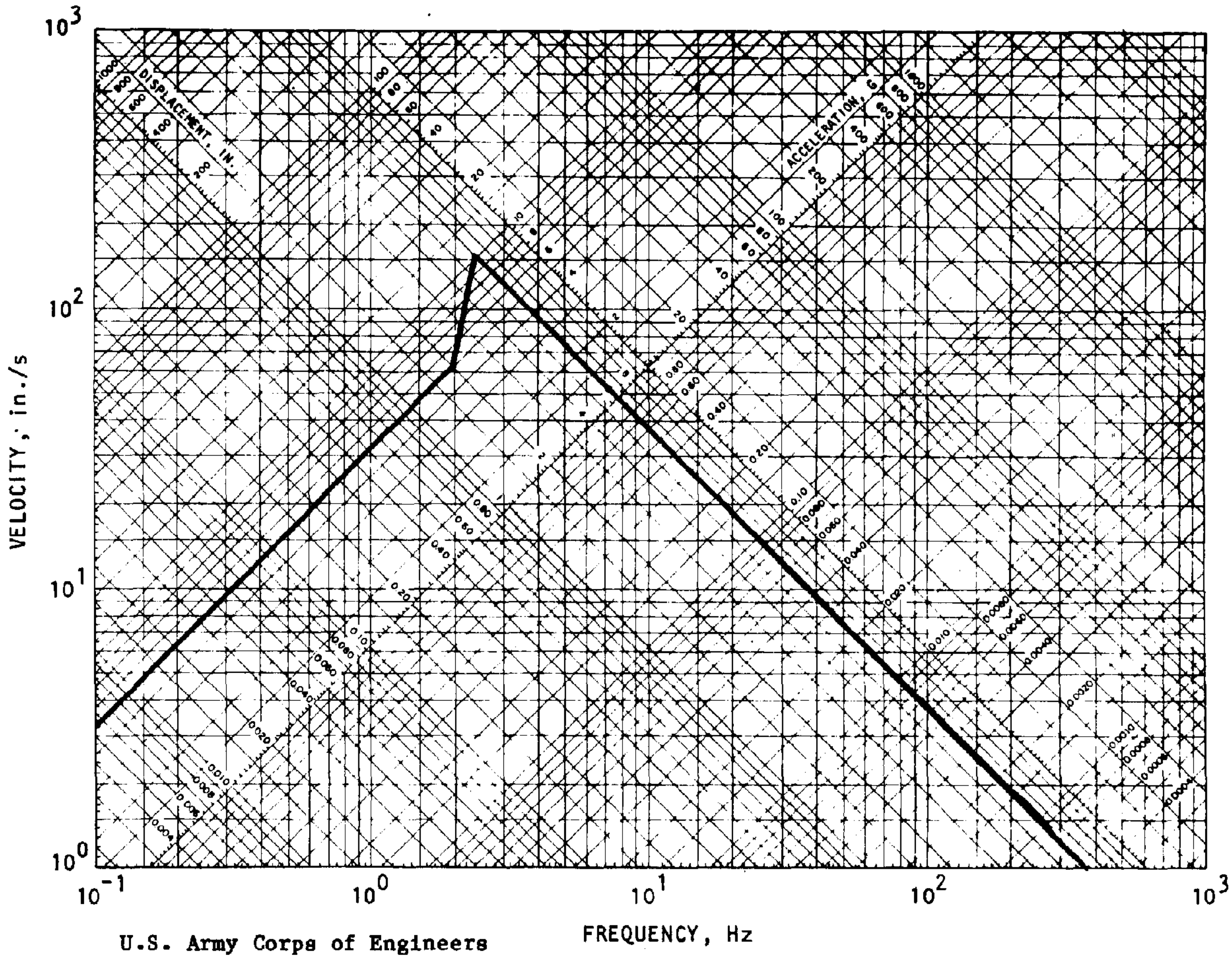

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FIGURE A-3L-1. HORIZONTAL SHOCK SPECTRUM, TRANSFORMER E16SS

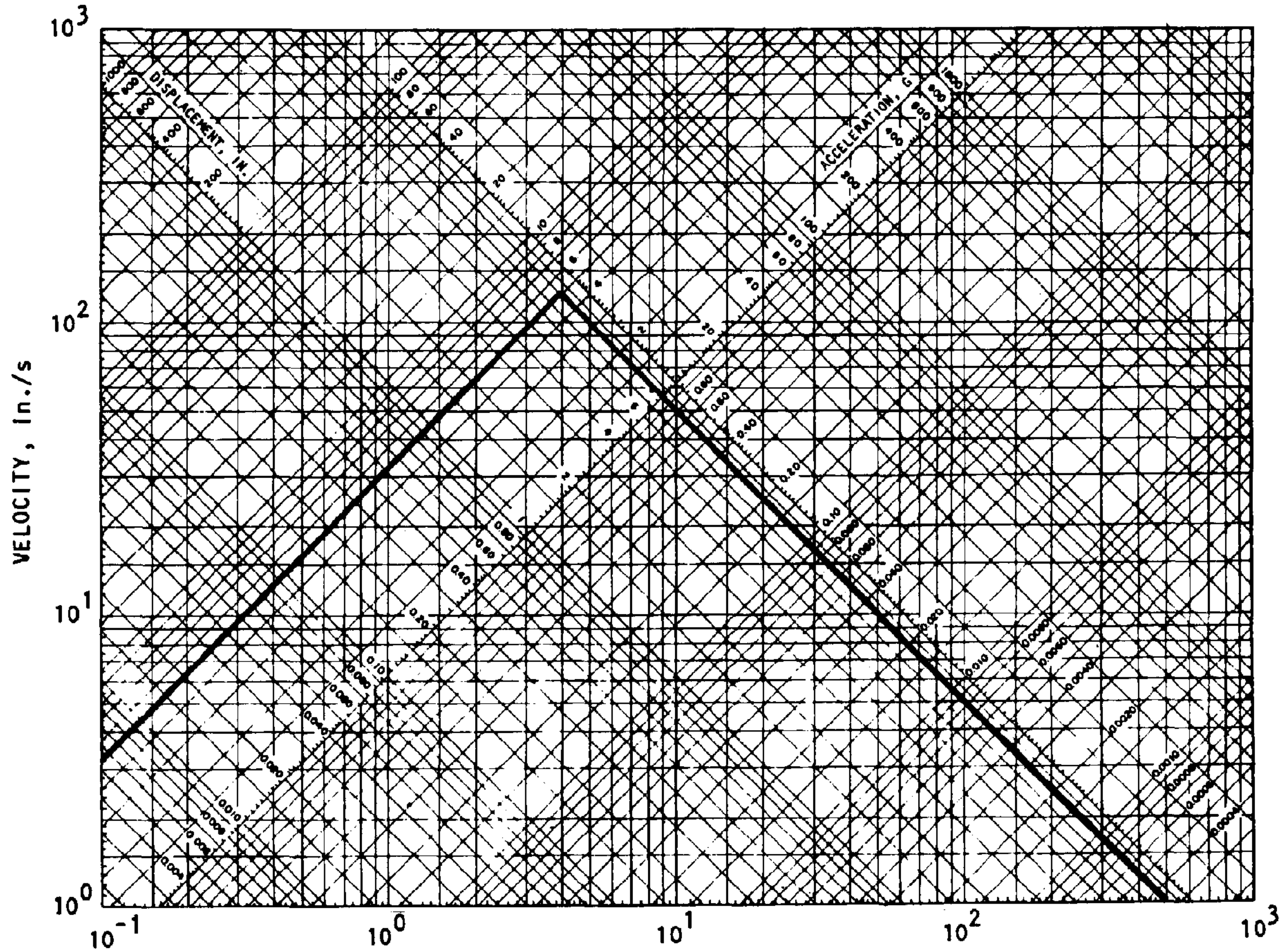

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FIGURE A-3L-2. VERTICAL SHOCK SPECTRUM, TRANSFORMER E16SS

A-3M. TRANSFORMER

1. Transformer. This specimen was a Westinghouse, Type DT3, 75 kVA, 480, −120/208 volt, 3-phase transformer. This unit is a standard industrial dry nonruggedized transformer. The specimen was approximately 31 in. wide, 20 in. deep, and 31 in. high, and weighs approximately 650 lb. The specimen was floor mounted.

2. Test.

a. Shock spectra.

Horizontal	Figure A-3M-1
Vertical	Figure A-3M-2

b. Single axis tests.

3. Results. Passed.

4. References and credits.

The Ralph M. Parsons Co. *NIKE-X Data Report: Shock Tests of Six Selected Items of Equipment*, NX-SE-165. Los Angeles, CA: Parsons, Feb 1969.

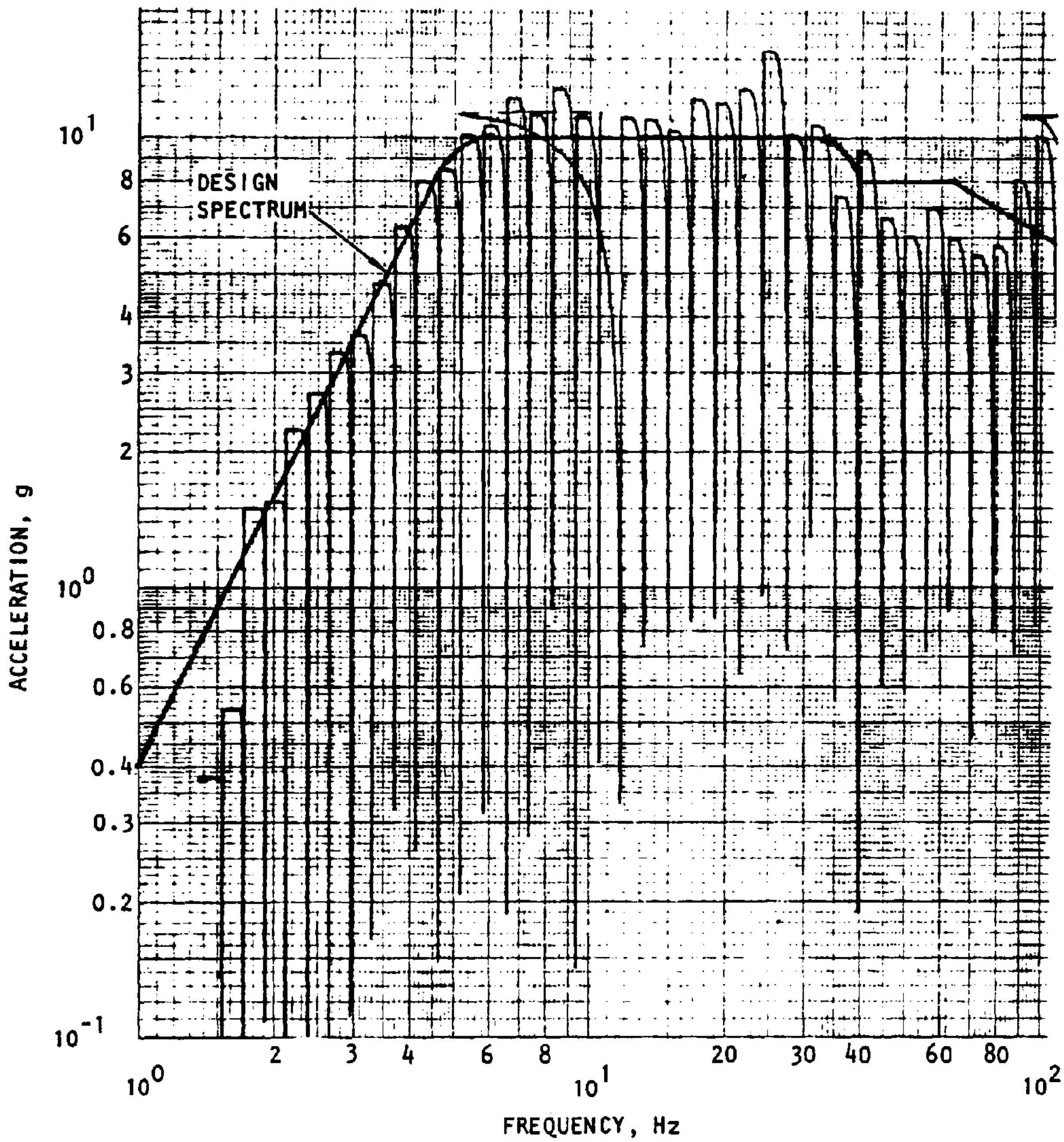

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FIGURE A-3M-1. COMPARISON OF DESIGN AND TEST RESPONSE SPECTRA, HORIZONTAL AXIS

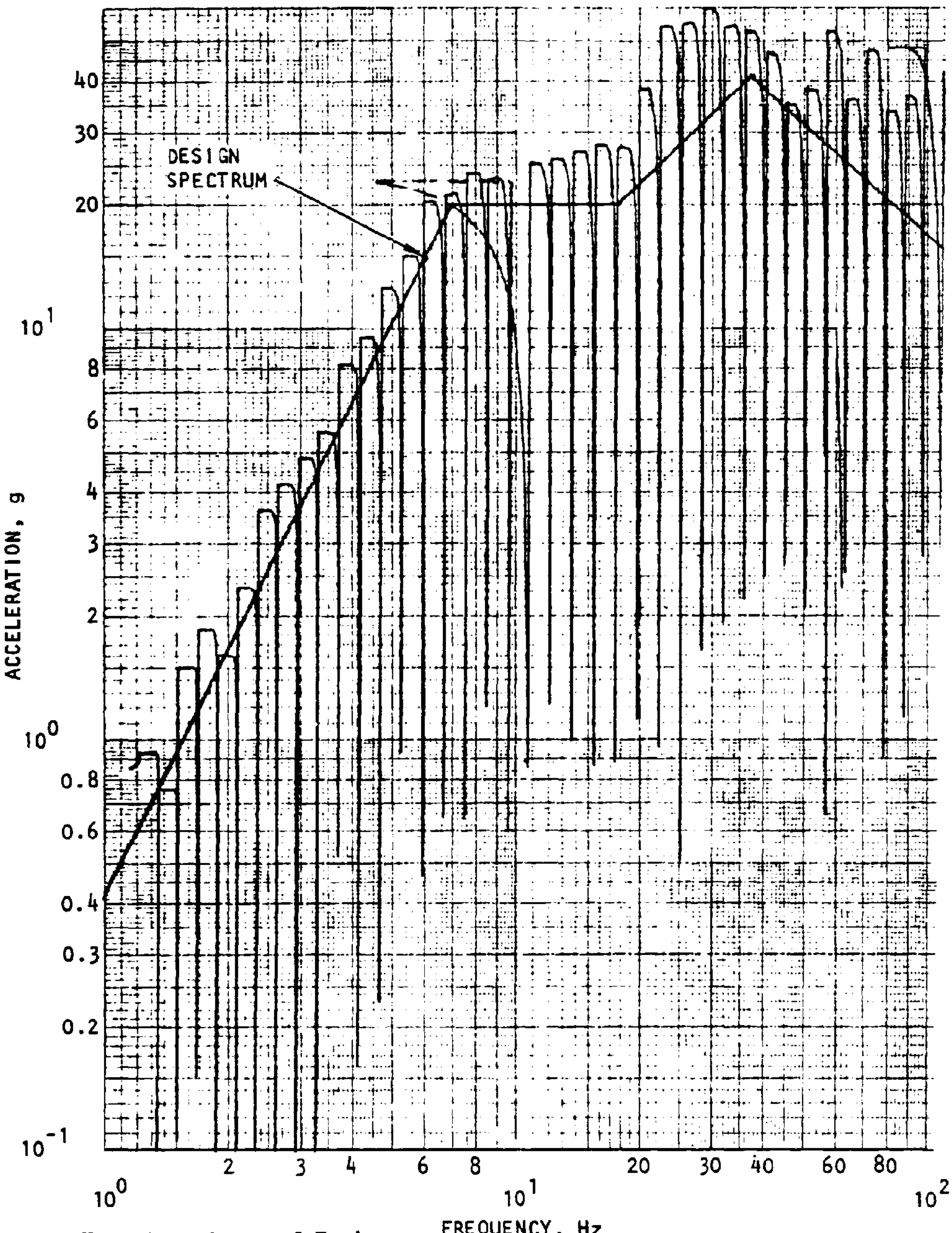

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FIGURE A-3M-2. COMPARISON OF DESIGN AND TEST RESPONSE SPECTRA VERTICAL AXIS

A-3N. DIESEL ENGINE COMPONENTS

1. Diesel engine M & C components.

a. Diesel engine control cabinet. Manufactured by Cooper Bessemer, weight 2000 lb, dimensions 103 in. by 60 in. by 50 in.

b. Generator neutral resistor. Manufactured by Euclid Electric, weight 750 lb, dimensions 29 in. by 51 in. by 48 in.

c. Fuel transfer pump. Manufactured by Roper, weight 100 lb, dimensions 48 in. by 36 in. by 12 in.

d. Fuel pump. Manufactured by Cooper Bessemer, weight 300 lb, dimensions 12 in. by 12 in. by 18 in.

e. Fuel pressure regulator. Manufactured by Cooper Bessemer, weight 50 lb, dimensions 18 in. by 8 in. by 8 in.

f. Overspeed governor. Manufactured by Woodward, weight 60 lb, dimensions 18 in. by 10 in. by 7 in.

g. Load sensing governor. Manufactured by Woodward, weight 150 lb, dimensions 30 in. by 12 in. by 8 in.

h. Variable timing actuator. Manufactured by Cooper Bessemer, weight 50 lb, dimensions 20 in. by 9 in. by 6 in.

i. Pressure transmitter. Manufactured by Bristol, weight 5 lb, dimensions 8 in. by 8 in. by 12 in.

j. Differential pressure transmitter. Manufactured by Johnson, weight 5 lb, dimensions 3 in. by 3 in. by 3 in.

2. Test.

a. Shock spectra.

(1) Diesel engine control cabinet (figs. A-3N-1 and A-3N-3)

(2) Generator neutral resistor (figs. A-3N-2 and A-3N-3)

(3) Fuel transfer pump (fig. A-3N-4)

(4) Fuel pump (fig. A-3N-4)

(5) Fuel pressure regulator (fig. A-3N-4)

(6) Overspeed governor (fig. A-3N-4)

(7) Load sensing governor (fig. A-3N-4)

(8) Variable timing actuator (fig. A-3N-4)

(9) Pressure transmitter (fig. A-3N-5)

(10) Differential pressure transmitter (fig. A-3N-6)

b. Single axis test. Two horizontal and vertical.

3. Results.

a. All test articles were subjected to 100 percent of the predicted shock level for the equipment. In addition, the test articles were in an operational and monitored mode during shock testing.

b. Results indicate that no functional degradation nor structural damage was observed during and after shock testing. All test articles were in an operational mode following shock testing with the exception of a thermocouple recorder in the diesel engine control console which became disabled due to unplugging of a connector. However, within minutes the connector was reinstalled and the recorder was operational.

4. References and credits.

Army Corps of Engineers (COE). *Test Report on Safeguard Shock Environmental Testing, Vol. I: Test Report*, HNDSP-74-324-ED-R. Huntsville, AL: COE, Dec 1974.

Army Corps of Engineers (COE). *Test Report on Safeguard Shock Environmental Testing, Vol. II: Test Procedures*, HNDSP-74-324-ED-R. Huntsville, AL: COE, Dec 1974.

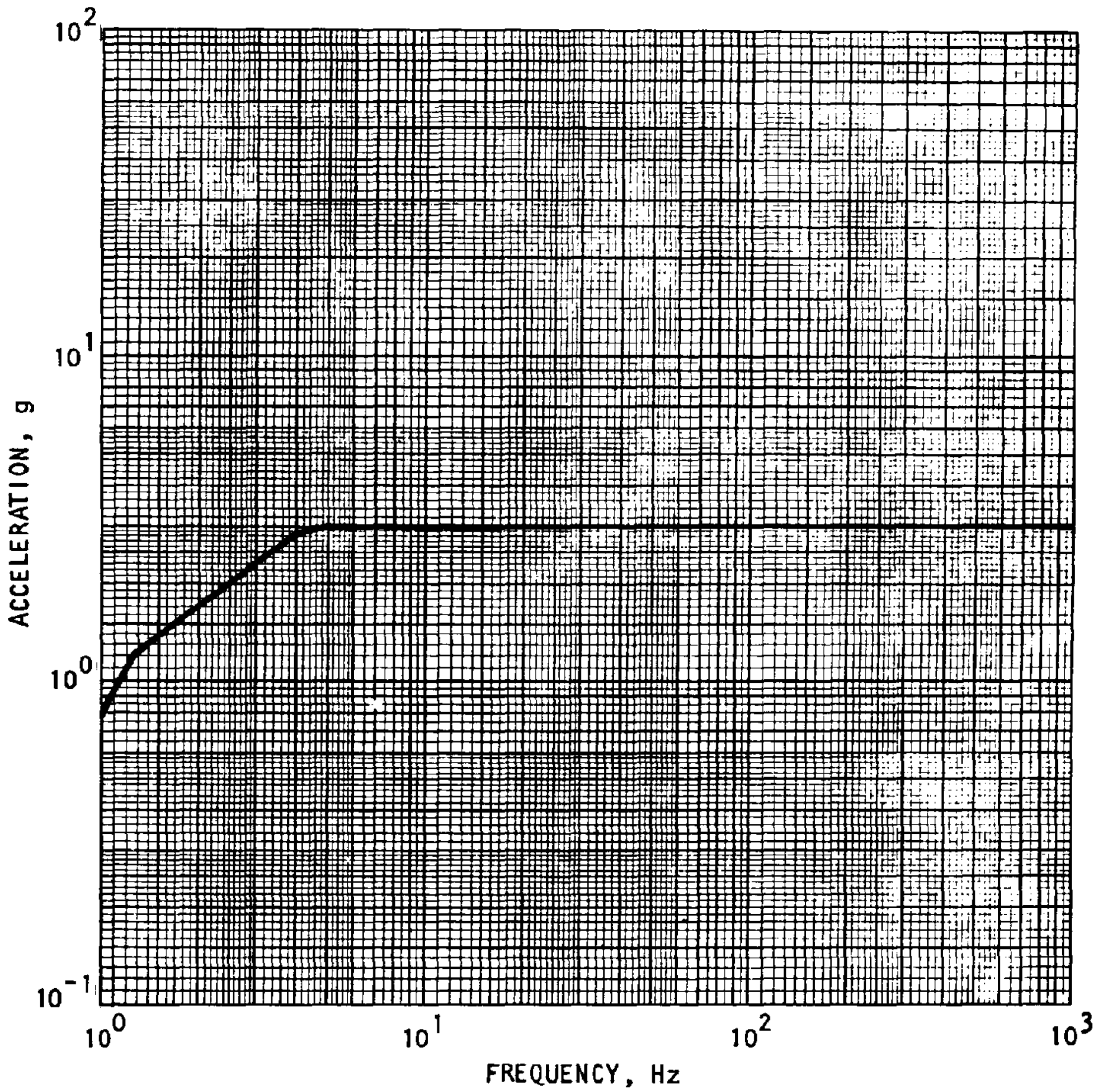

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FIGURE A-3N-1. SHOCK SPECTRUM NO. 1 (SS1) — USE FOR DIESEL ENGINE CONTROL — HORIZONTAL

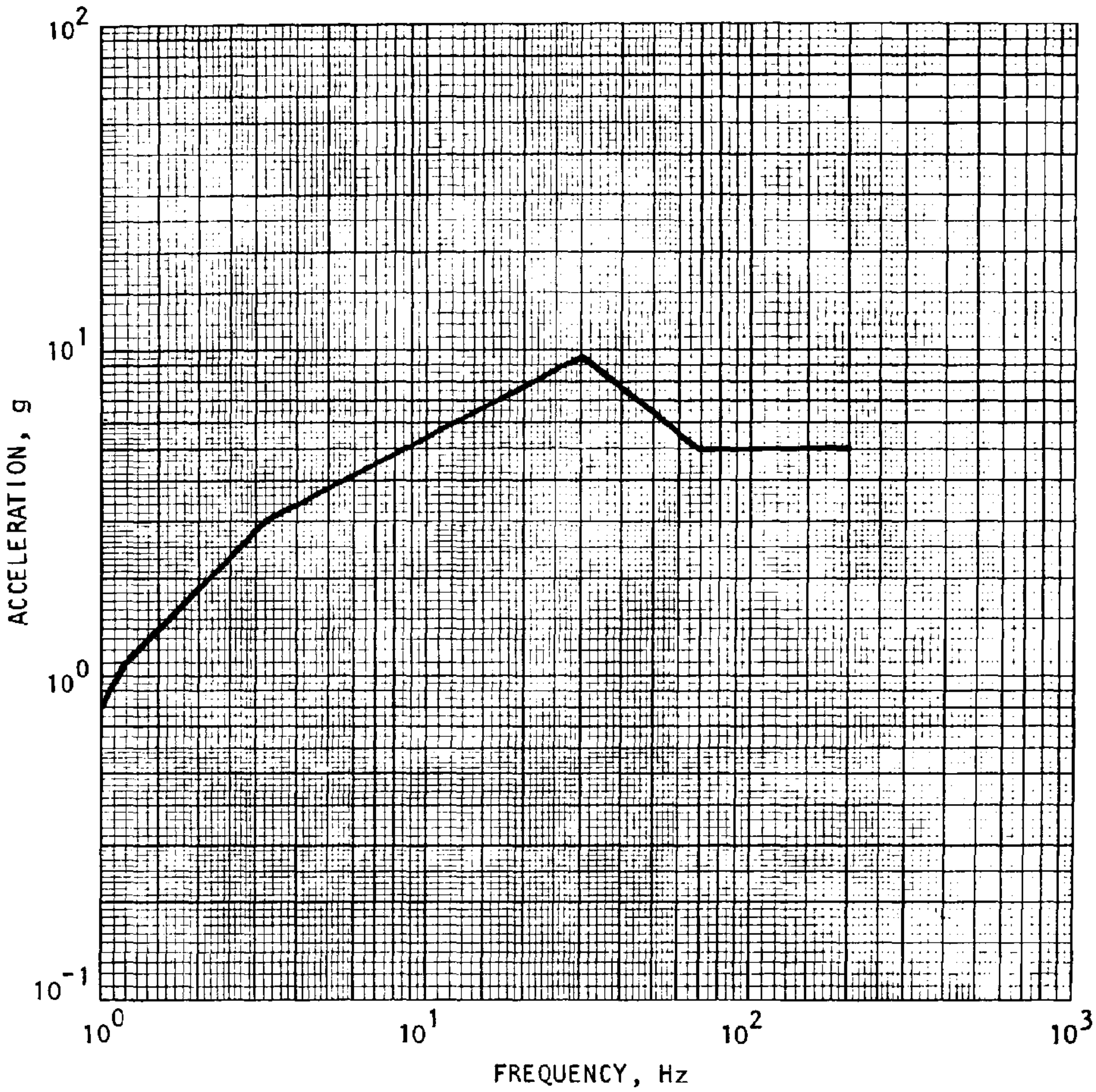

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FIGURE A-3N-2. SHOCK SPECTRUM NO. 2 (SS2) — USE FOR GENERATOR NEUTRAL RESISTOR — HORIZONTAL

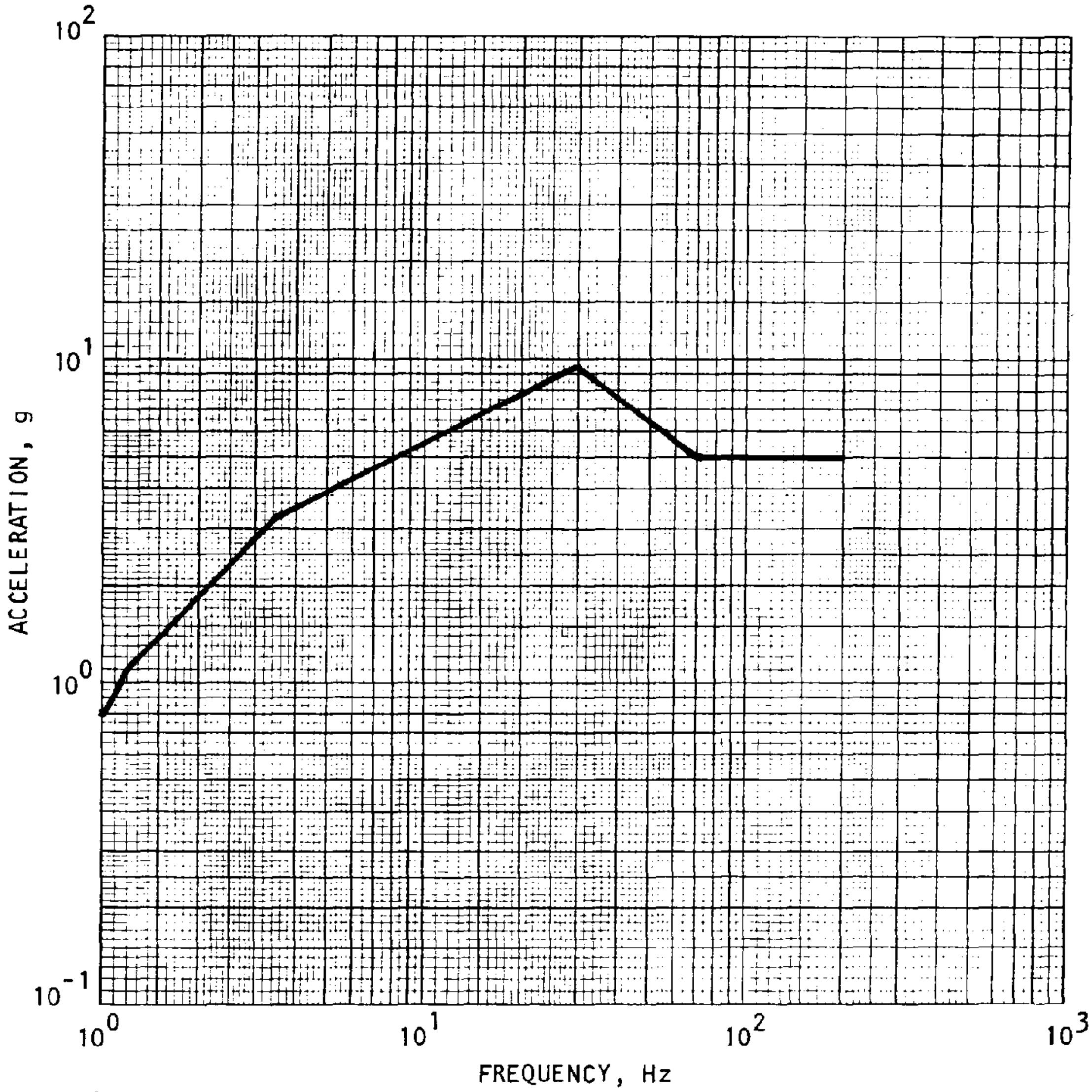

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FIGURE A-3N-3. SHOCK SPECTRUM NO. 3 (SS3) — USE FOR DIESEL ENGINE CONTROL — VERTICAL, GENERATOR NEUTRAL RESISTOR — VERTICAL

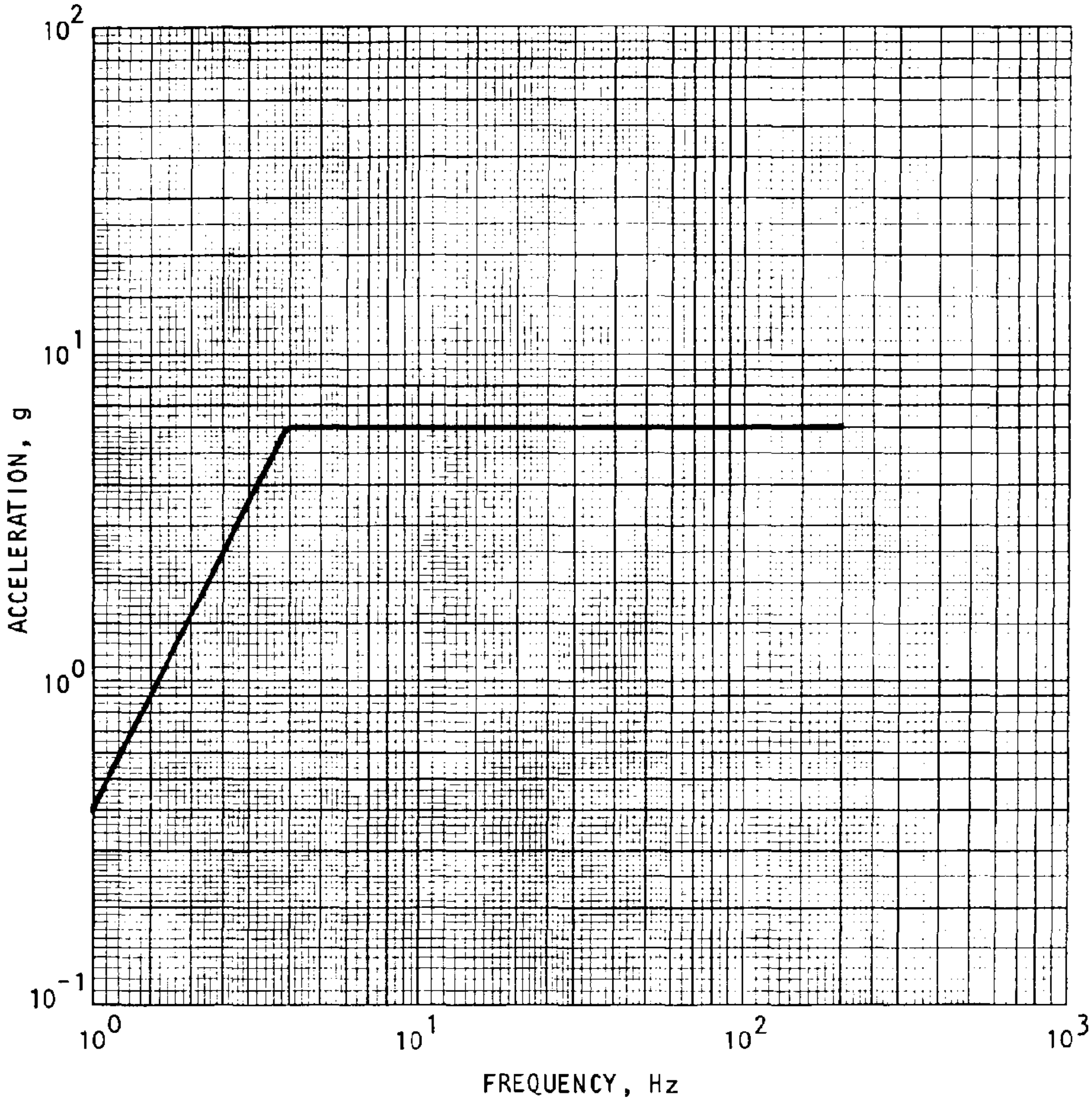

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FIGURE A-3N-4. SHOCK SPECTRUM NO. 4 (SS4) — USE FOR FUEL TRANSFER PUMP, FUEL PUMP, FUEL PRESSURE REGULATOR, OVERSPEED GOVERNOR AND VARIABLE TIMING ACTUATOR BOTH VERTICAL AND HORIZONTAL

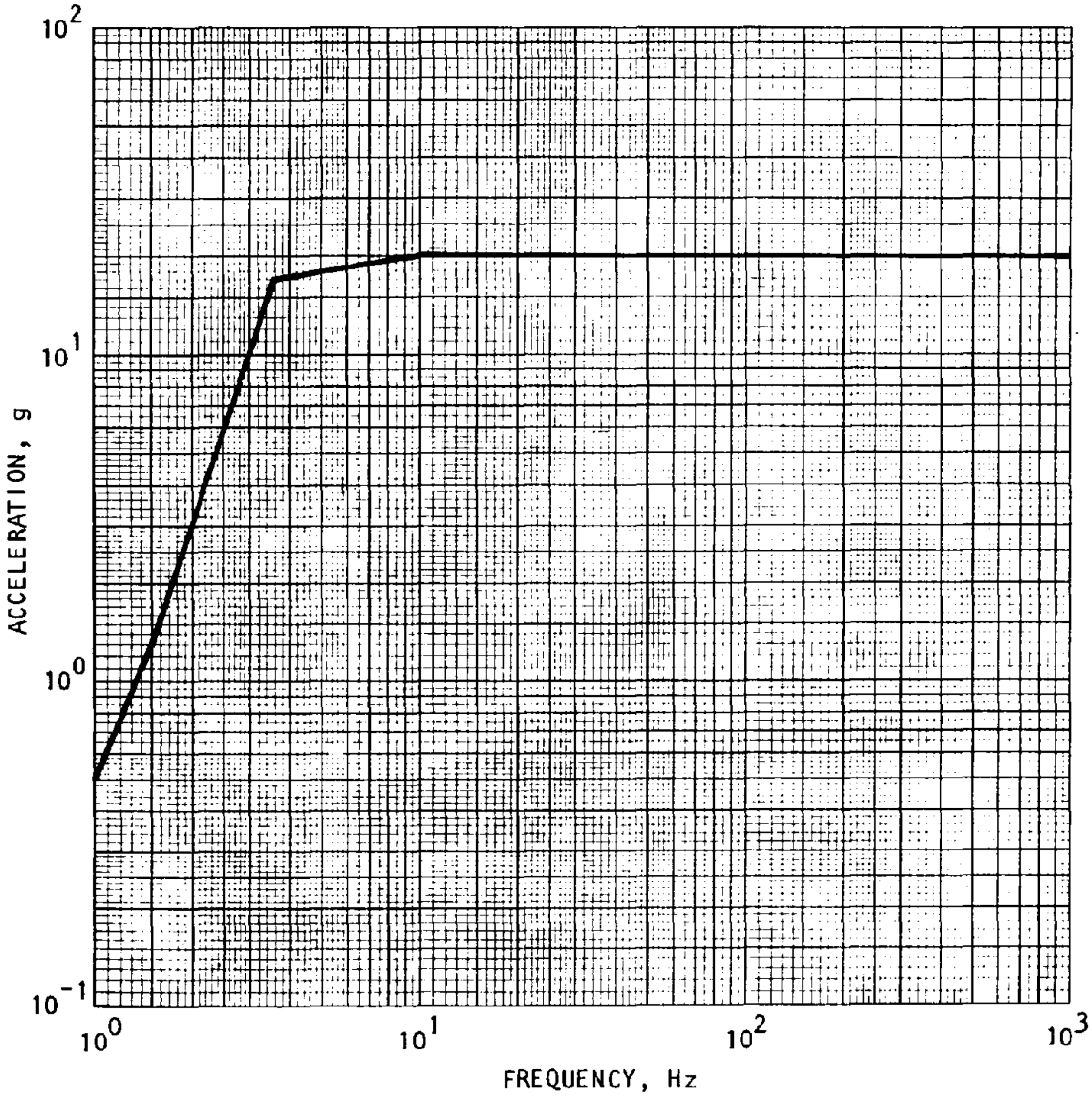

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FIGURE A-3N-5. SHOCK SPECTRUM NO. 5 (SS5) — USE FOR PRESSURE TRANSMITTER AND DIFFERENTIAL PRESSURE TRANSMITTER — HORIZONTAL

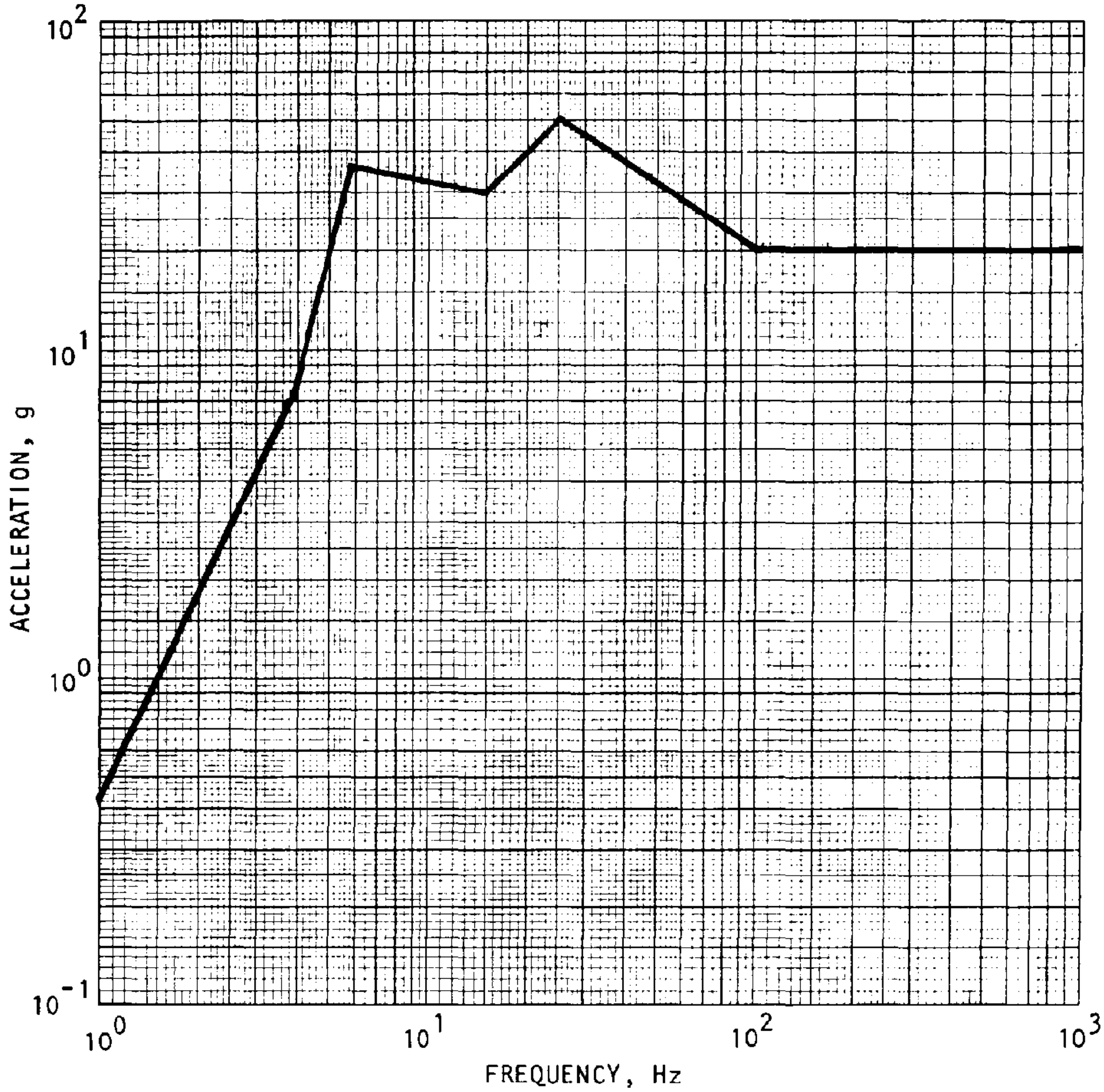

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FIGURE A-3N-6. SHOCK SPECTRUM NO. 6 (SS6) — USE FOR PRESSURE TRANSMITTER AND DIFFERENTIAL PRESSURE TRANSMITTER — VERTICAL

A-30. MOTOR STARTER AND SURGE PAK

1. Motor starter and surge pak cabinets.

Made by Westinghouse Electric Corporation. The starter cabinet is 6 ft 4 in. wide, 8 ft 4 in. high, and 2 ft 4 in. deep. The surge pak cabinet is 3 ft 6 in. wide, 6 ft 6 in. high, and 2 ft 11 in. deep. The exciter/regulator cabinet was made by Basler Electric Company and is 5 ft wide, 7 ft 6 in. high, and 4 ft 2 in. deep. Starter and exciter cabinets weigh 7000 lb and surge pack cabinet weighs 1271 lb.

2. Test.

a. Shock spectra.

Starter and exciter (fig. A-30-1 and A-30-2)
Surge pak (figs. A-30-3 and A-30-4)

b. Biaxial test.

3. Tests. Damage to the starter cabinet was limited to breaking of the door latches and hinges during tests 20, a 100 percent X-Z axis test; 27, a 50 percent Y-Z axis test; 29 and 30, both 75 percent Y-Z axis tests; and 31 and 35, both 100 percent Y-Z axis tests. This failure could cause additional jarring of the relay contacts mounted on the door. Since the wires connecting these contacts to the cabinet are routed along the hinge, no damage to the wires should occur unless the hinge breaks. Cutting these wires would have the same effect on the equipment as deleting fault protection from the system, with the additional possibility of fire, due to shorting of live wires to the cabinet frame. Throughout most of the series of during-shock tests, the seal-in units and/or indicating contactor switches of protective relays experienced chatter of a duration adequate to initiate breaker tripping. Motor starter cabinet became very loose.

4. References and credits.

Army Corps of Engineers (COE). *Test Report on Safeguard Shock Environment Test of Motor-Generator Set E12GM*, HNDSP-74-338-ED-R. Huntsville, AL: COE, Dec. 1974.

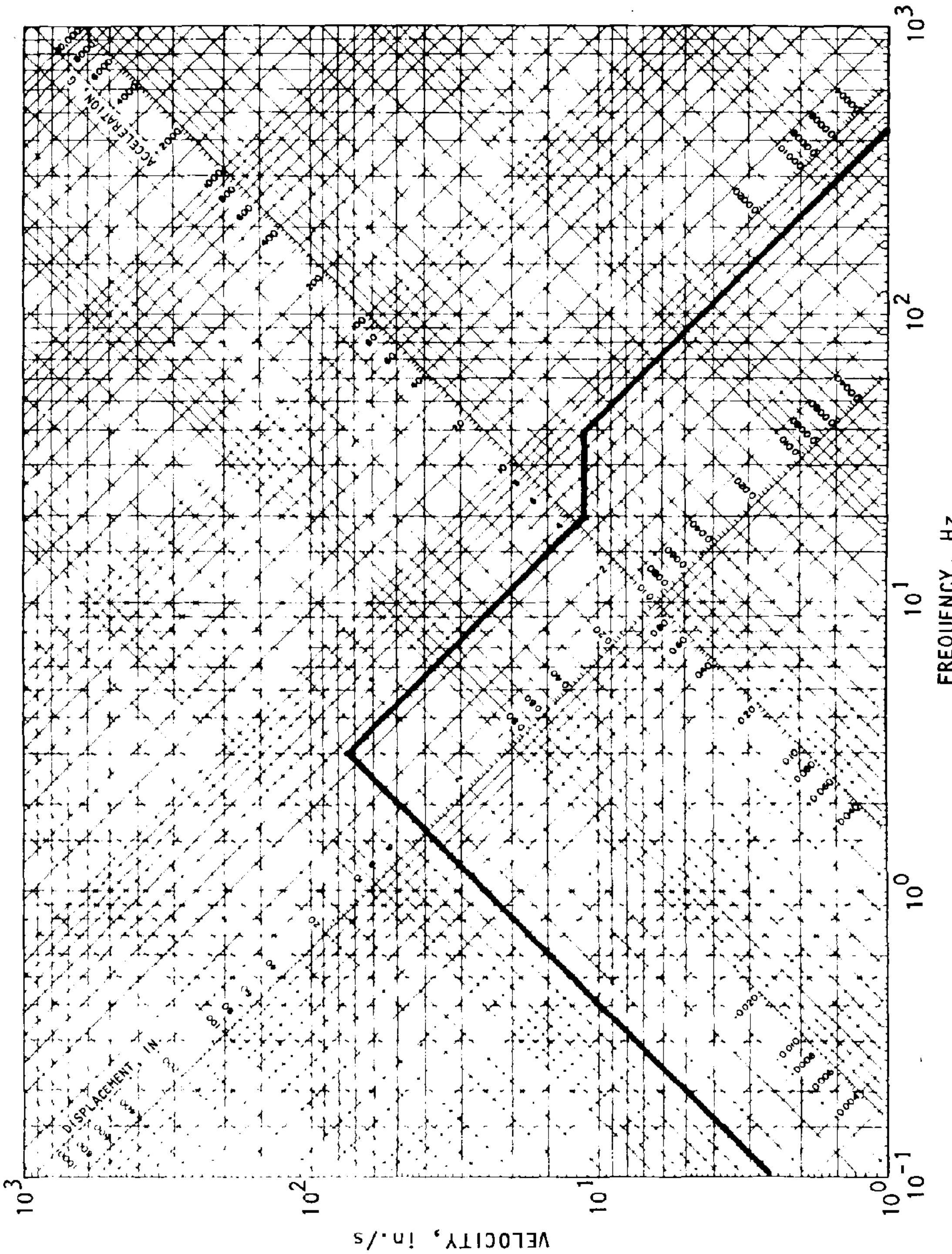

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FIGURE A-30-1. VERTICAL SHOCK SPECTRUM FOR STARTER AND EXCITER CABINETS

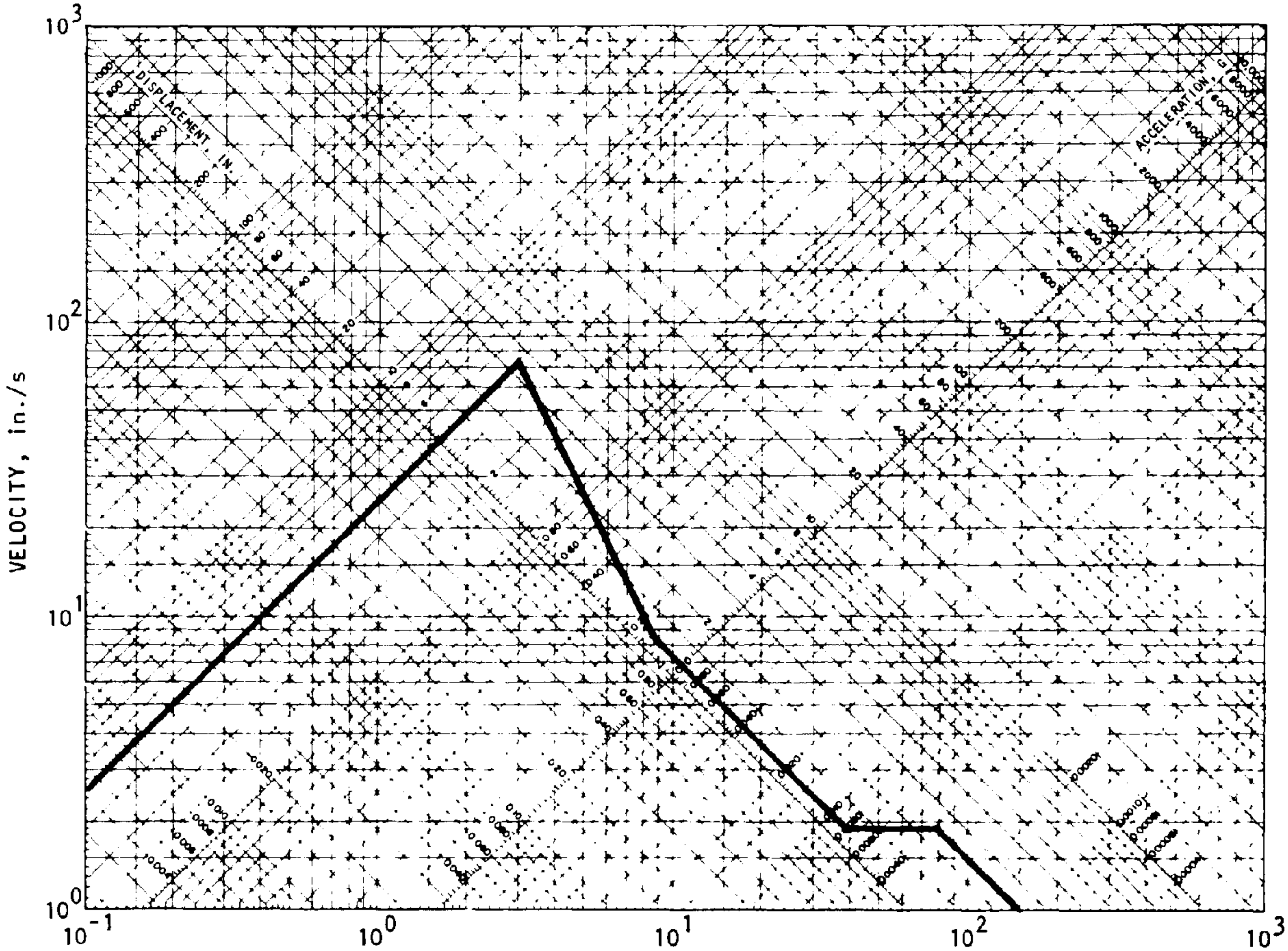

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FIGURE A-30-2. HORIZONTAL SHOCK SPECTRUM FOR STARTER AND EXCITER CABINETS

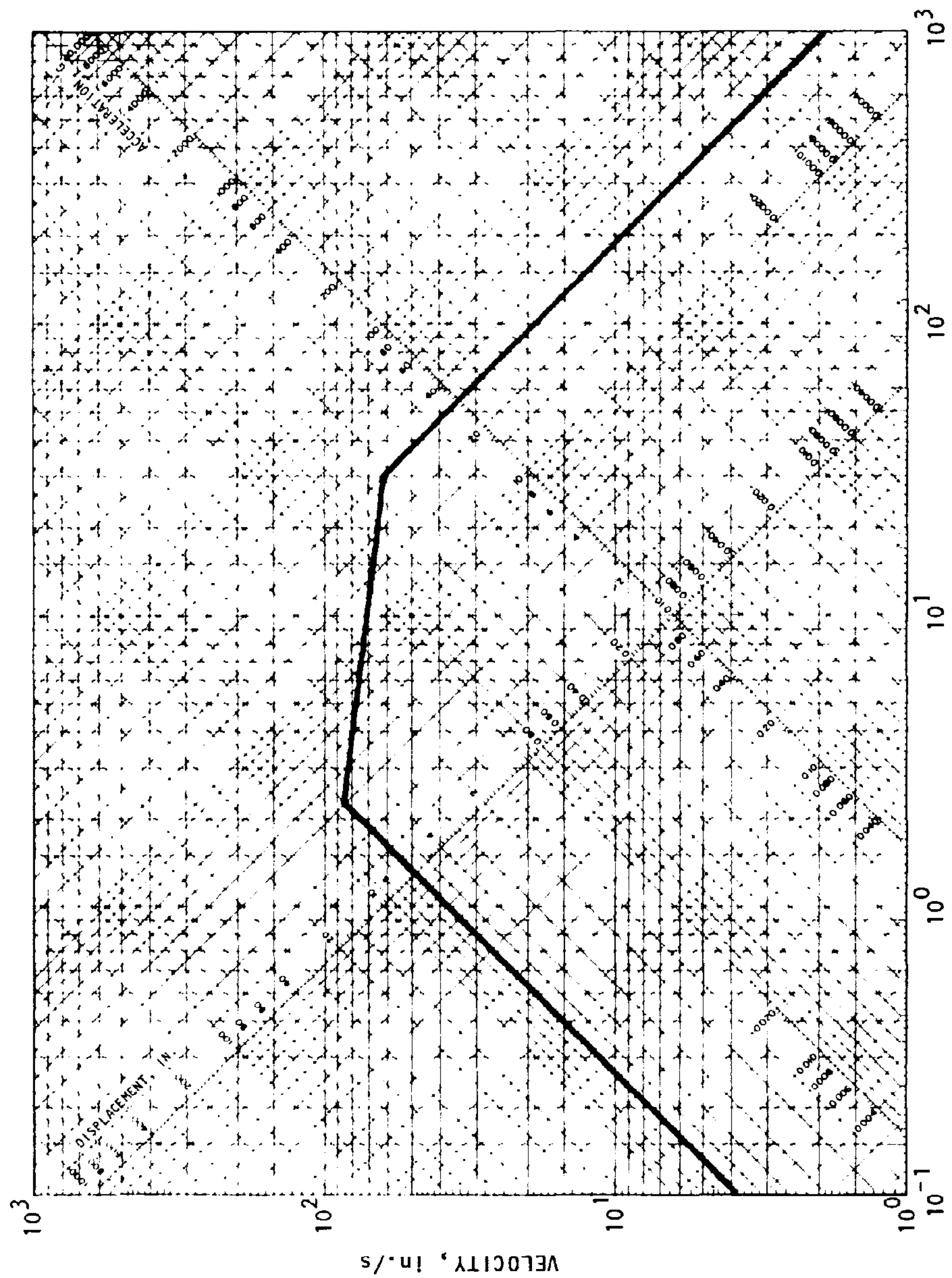

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FIGURE A-30-3. VERTICAL SHOCK SPECTRUM FOR SURGE PAK CABINET

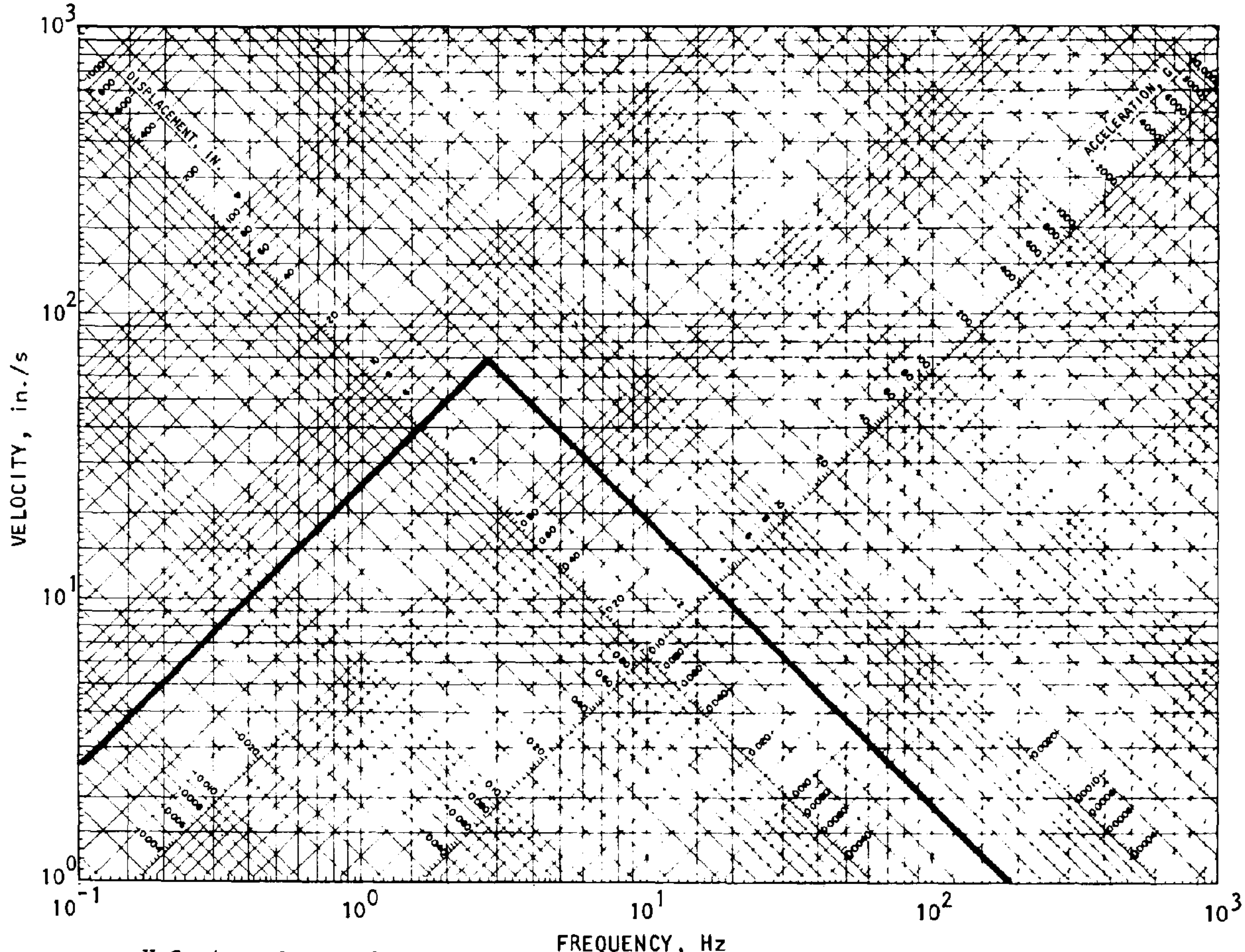

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FIGURE A-30-4. HORIZONTAL SHOCK SPECTRUM FOR SURGE PAK CABINET

A-3P. MONITOR AND CONTROL COMPONENTS

1. Monitoring and control components.

Nomenclature	Weight lb	Space Envelope, in. L	by W	by H	Manufacturer
P&ID outside air makeup	1065	36	24	90	Fairchild Hiller
Local instrument	210	36	24	15	Fairchild Hiller
Local instrument	210	37	25	12	Fairchild Hiller
Pressure switch	34	17	14	6	Foxboro
Pressure switch	1	3.5	2.5	7	Auen-Bradley
Pneumatic control valve	206	28	16	22	Honeywell
Pneumatic actuator	75	37	18	13	Masoneilan
Butterfly valve	700	58	35	31	Cover Corp.
Indicating control assy	44	16	10	14	Honeywell
Butterfly valve	77	45	13	25	Dover Corp.
Plug valve	197	22	18	24	Rockwell
Plug valve	245	33	10	30	Tuflite
Temperature transmitter	1.5	3	4	6.5	Consolidated Dev.
Plug valve	6200	72	36	56	Rockwell

2. Test.

a. Shock spectra.

ITC No./Nomenclature		Figure No.
163PL	P&ID Outside air makeup	A-3P-1, A-3P-2
150PL	Local instrument	A-3P-1, A-3P-2
153PL	Local instrument	A-3P-1, A-3P-2
154AP	Pressure switch	A-3P-1, A-3P-2
003SD	Pressure switch	A-3P-1, A-3P-2
P83CV	Pneumatic control valve	A-3P-1, A-3P-2
I18DA	Pneumatic actuator	A-3P-2, A-3P-4
P12VQ	Butterfly valve	A-3P-3, A-3P-5
I03CL	Indicating control assy	A-3P-3, A-3P-5
P22VQ	Butterfly valve	A-3P-3, A-3P-5
P05VJ	Plug valve	A-3P-3, A-3P-5
P67VJ	Plug valve	A-3P-3, A-3P-4
I01CV	Temperature transmitter	A-3P-2, A-3P-4
P65VJ	Plug valve	A-3P-3, A-3P-5

b. Biaxial test and single axis test. For items outside air makeup, pneumatic actuator, butterfly valve, and plug valve.

3. Results.

a. Structural degradation and one functional anomaly occurred during test of the outside air makeup panel: A swedge lock swivel in the panel sheared off at the point it enters the differential pressure transmitter; a brace was added for support and the specimen was retested with no recurrence of the failure; during testing a front panel fell off and the lower framework of the cabinet cracked; however, the structural integrity of the cabinet or operation of the specimen was not affected; the pneumatic receiver indicating controller malfunctioned after the second 100 percent level shock in the Y axis and the output pressure shut off below the control index point. However, two similar units were tested on local instrument panel 150PL and no malfunctions were encountered.

b. No major functional degradation nor structural damage occurred on the remaining 13 items tested.

4. References and credits.

Army Corps of Engineers (COE). *Test Report on Safeguard Shock Environment of Monitor and Control Components for U.S. Army, Engineer Division, Huntsville, Corps of Engineers*, HNDSP-74-320-ED-R. Huntsville, AL: COE, Dec 1974.

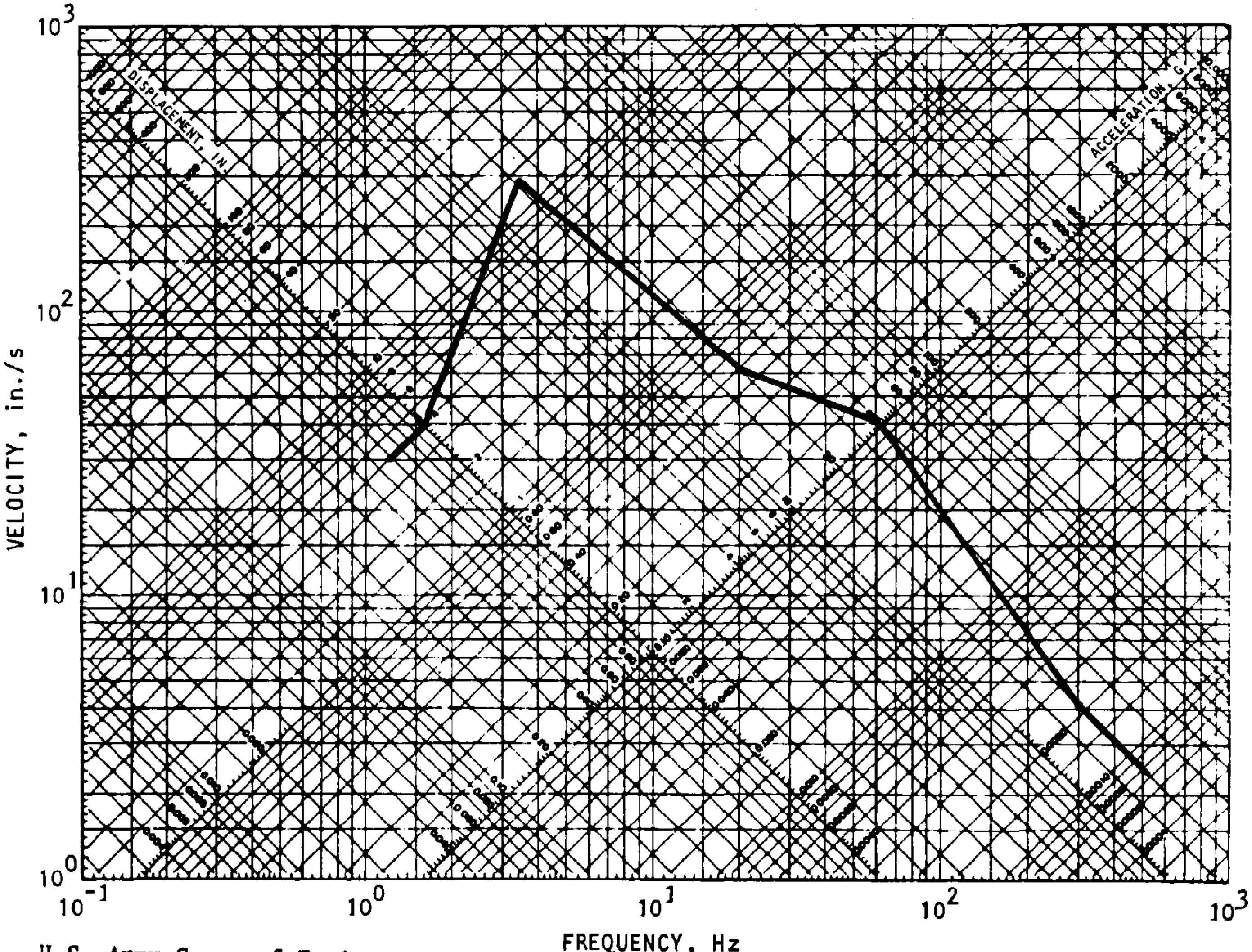

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FIGURE A-3P-1. HORIZONTAL SHOCK SPECTRUM FOR OUTSIDE AIR MAKE-UP, LOCAL INSTRUMENTS, PRESSURE SWITCHES, AND PNEUMATIC CONTROL VALVE

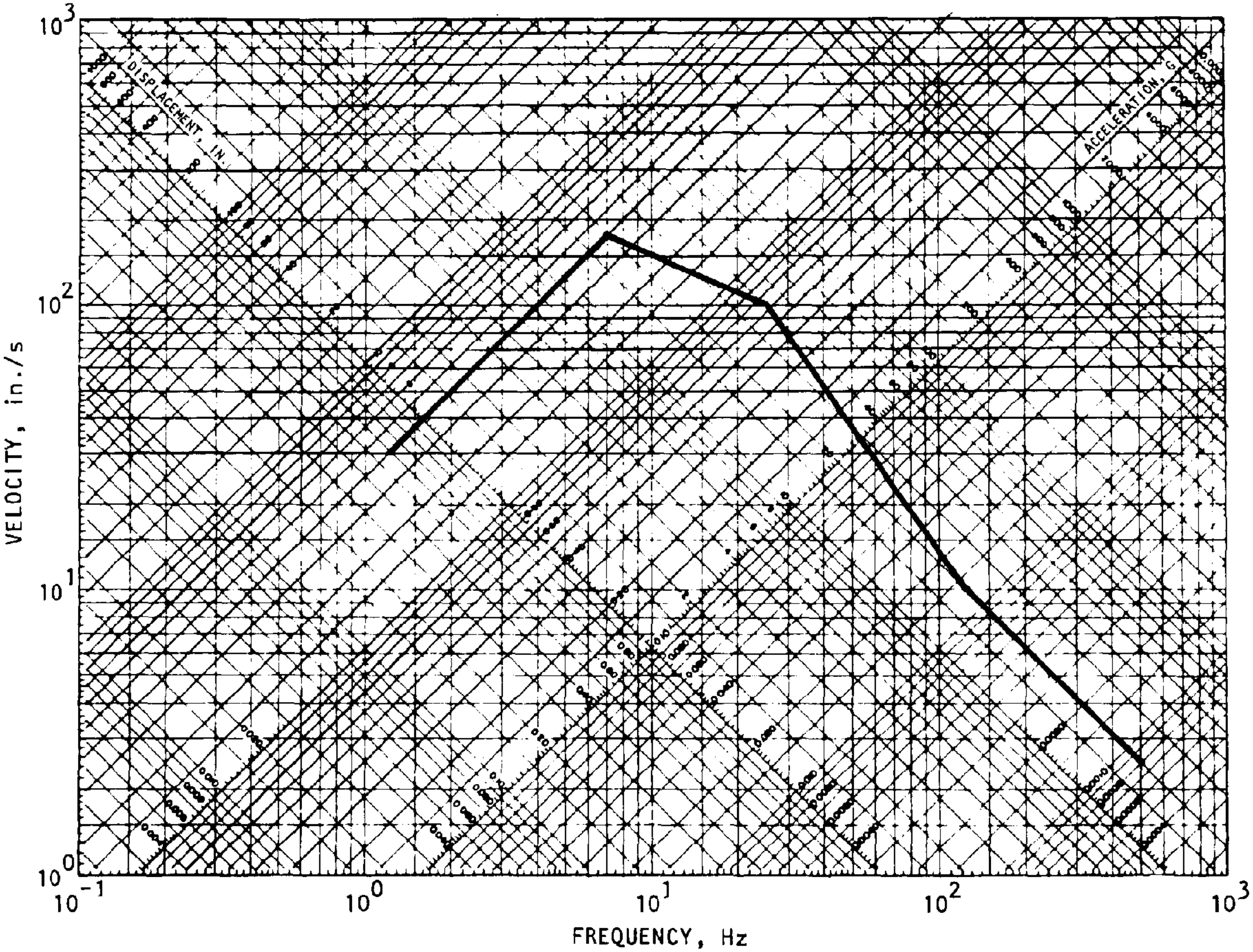

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FIGURE A-3P-2. VERTICAL SHOCK SPECTRUM FOR OUTSIDE AIR MAKE-UP, LOCAL INSTRUMENTS, PRESSURE SWITCHES, PNEUMATIC CONTROL VALVE, PNEUMATIC ACTUATOR, AND TEMPERATURE TRANSMITTER

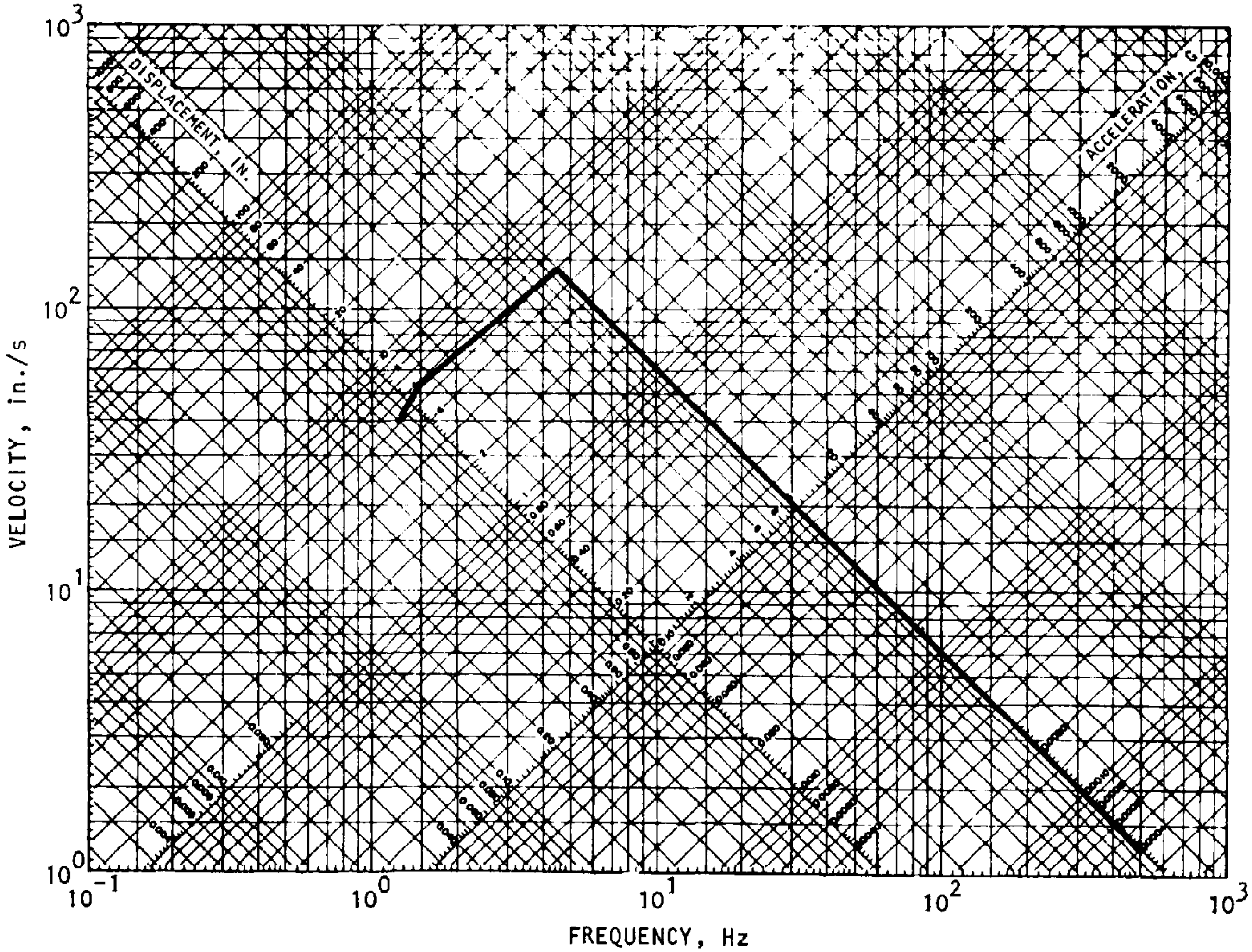

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FIGURE A-3P-3. HORIZONTAL SHOCK SPECTRUM FOR BUTTERFLY VALVES, PLUG VALVES, AND INDICATING CONTROL ASSEMBLY

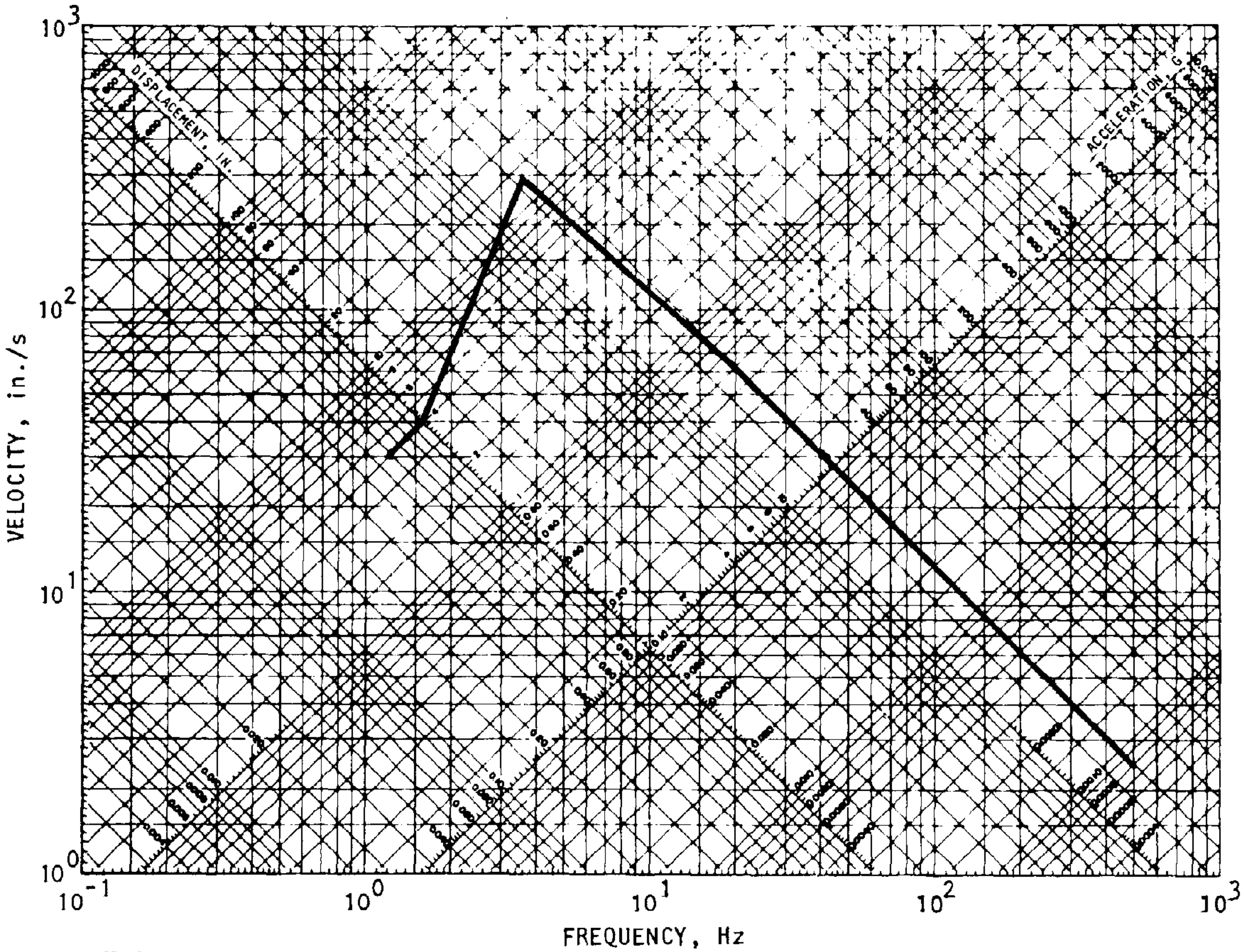

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FIGURE A-3P-4. HORIZONTAL SHOCK SPECTRUM FOR PNEUMATIC ACTUATOR, PLUG VALVE, AND TEMPERATURE TRANSMITTER

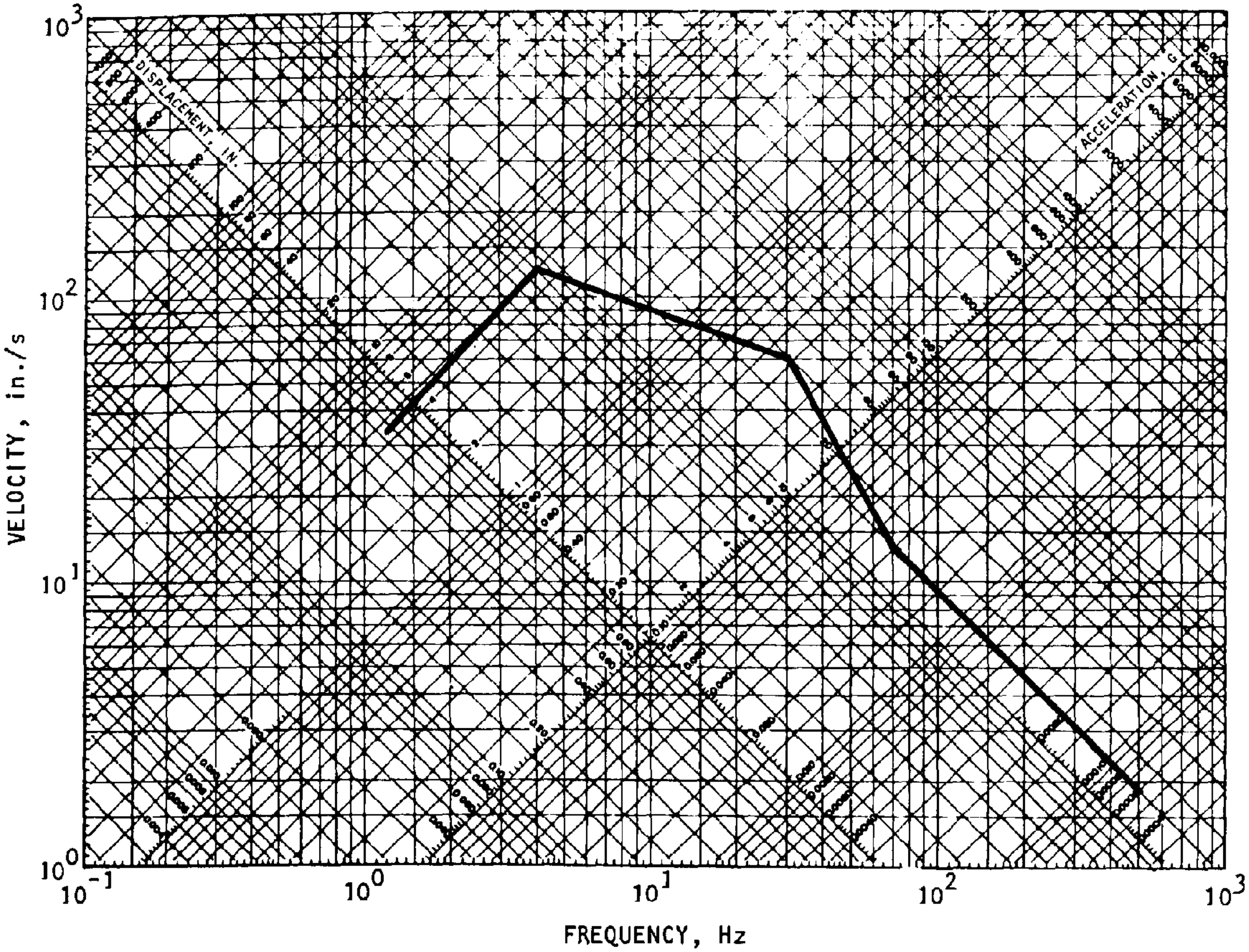

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FIGURE A-3P-5. VERTICAL SHOCK SPECTRUM FOR BUTTERFLY VALVES, PLUG VALVES, AND INDICATING CONTROL ASSEMBLY

A-3Q. RELAY PANEL

1. Relay panel. The specimen was a Westinghouse, NEMA Type 1 panel enclosure, wall mounted. It was equipped with four off-on pushbuttons and four pilot lamps wired independently to four relays. The relays were the type used for machine tools and were of various load ratings.

2. Test.

a. Shock spectra.

Horizontal	Figure A-3Q-1
Vertical	Figure A-3Q-2

b. Single axis tests.

3. Results. Passed.

4. References and credits.

The Ralph M. Parsons Co. *NIKE-X Data Report: Shock Tests of Six Selected Items of Equipment*, NX-SE-165. Los Angeles, CA: Parsons, Feb 1969.

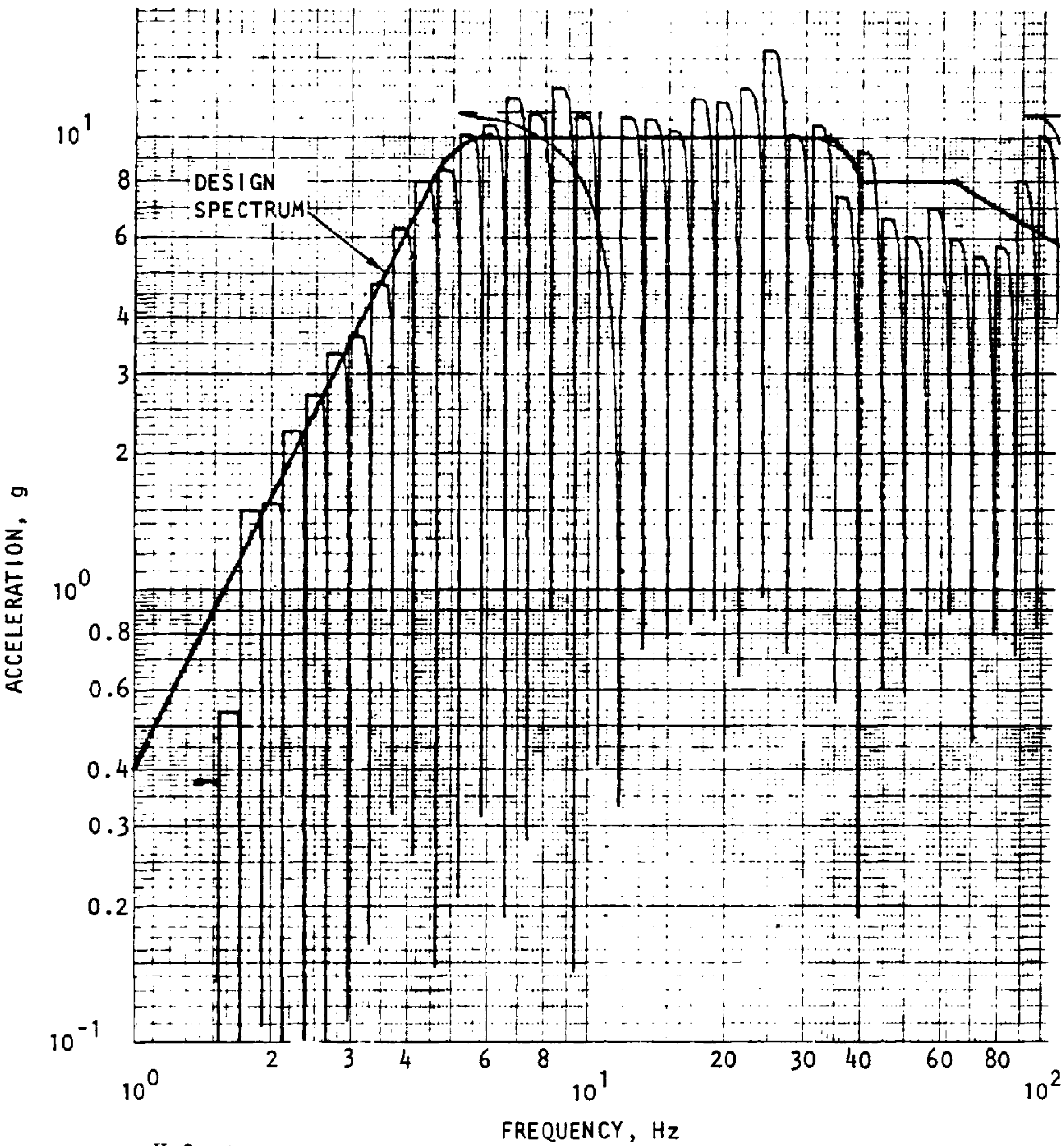

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FIGURE A-3Q-1. COMPARISON OF DESIGN AND TEST RESPONSE SPECTRA, HORIZONTAL AXIS

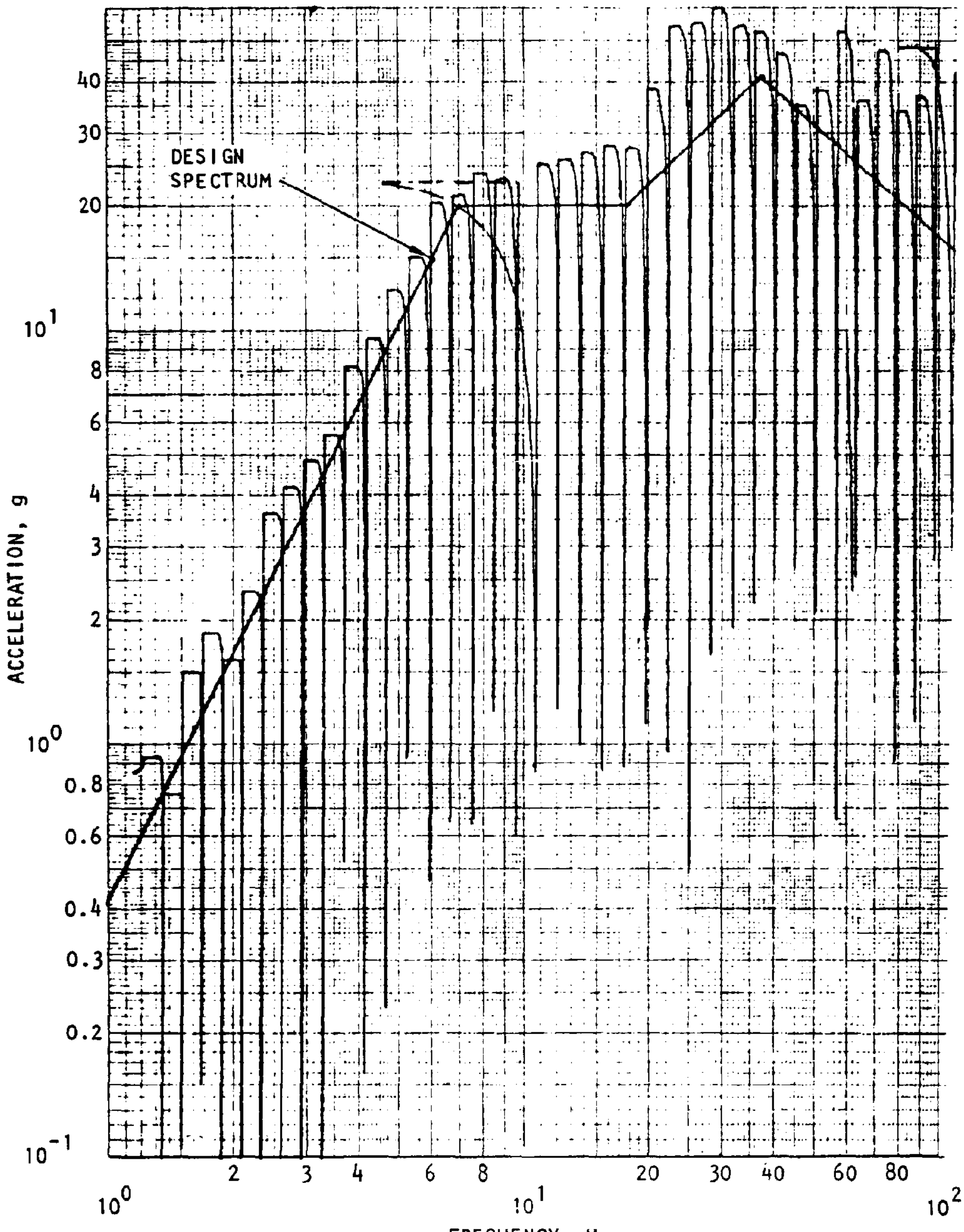

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FIGURE A-3Q-2. COMPARISON OF DESIGN AND TEST RESPONSE SPECTRA, VERTICAL AXIS

A-3R. GAS TURBINE/AC GENERATOR

1. Gas turbo-generator assembly.

a. The test specimen consisted of a gas turbine, gear reducer, and three-phase AC generator, mounted on a common base frame. Total weight of the equipment is 6000 lb.

b. The gas turbine assembly is an AiResearch Company Model 831-500 gas turbine, serial number (S/N) P-125, with an AiResearch gear reducer, 24:1 ratio, Model GBS831500, S/N P-101, driving an Electric Machinery Co. generator, S/N 271177131. Rated output of the unit is 400 kW, 277/480 V, 3 phase, 60 Hz, at 1800 rpm. The assembly is 11 ft 9 in. long 4 ft wide, and 4 ft high.

c. The engine control assembly was contained in a box mounted on the turbine base frame and thus was subjected to shock motions.

2. Test.

a. Shock spectra.

Horizontal and vertical	Figure A-3R-1

b. Biaxial test.

3. Results. The turbine-generator was successfully tested (no failures) on both axes with no damage. An 8-hr postshock run at 300 kW load was performed with no degradation from the established data base.

4. References and credits.

Army Corps of Engineers (COE). *Test Report on Safeguard Shock Environment Test of Gas Turbine Electric Generator E01GT,* HNDSP-74-329-ED-R. Huntsville, AL: COE, Dec 1974.

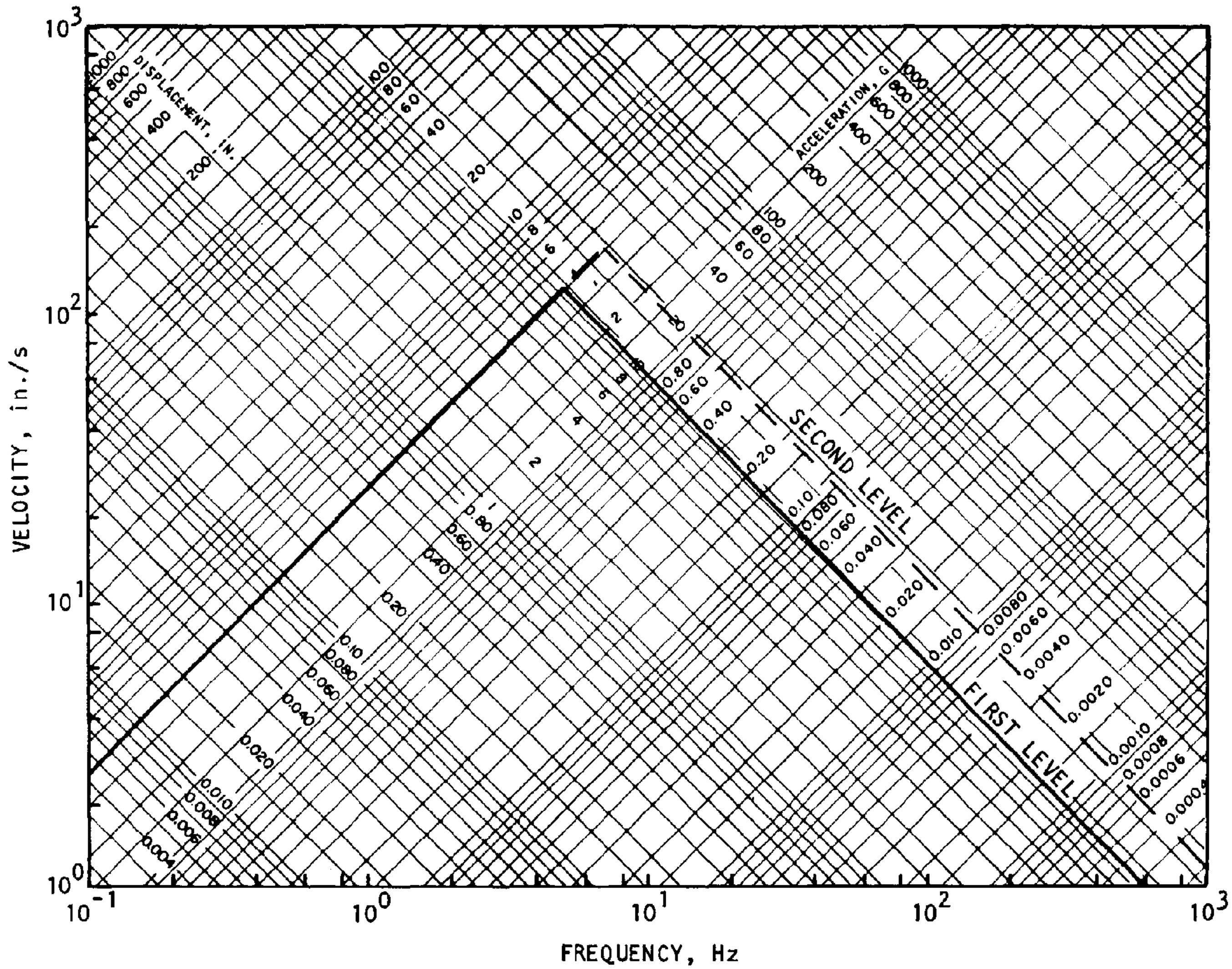

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FIGURE A-3R-1. SHOCK SPECTRA — HORIZONTAL AND VERTICAL

A–3S. CHILLER

1. 660-ton chiller components. The test specimens consisted of a control console, oil pump, relief valve, and oil cooler. The control console was tested as one test package and the oil pump, relief valve and oil cooler as another package.

DESCRIPTION OF TEST SPECIMEN

Item Type	*Weight, lb*	*Space Envelope, in.*	*Manufacturer*
Oil Pump and Relief Valve	60	19 by 15 dia.	Century
Oil Cooler	45	30 by 7 by 7	Century
Control Console	800	66 by 27 by 76	The Trane Co.

2. Test.

a. Shock spectra.

Horizontal	Figure A-3S-1
Vertical	Figure A-3S-2

b. Single axis test.

3. Results.

a. Pressure electric switch (PE-1). During shock applications nos. 1, 2, 3, and 4, X axis, the pressure electric switch momentarily opened. This caused the TR-1 and TR-3 to recycle and the load limiting relay (LLR) to drop out until the TR-3 completed its timing cycle. Corrective action:

> Testing was continued after TR-3 completed cycle. The pressure electric switch is a mercury filled glass bulb type and is sensitive to movement in the X axis direction.

b. Timing relay (TR-3). There was no functional degradation noted prior to, during, or after actual shock applications. However, due to momentary opening of the pressure electric switch (PE-1) the timing relay (TR-3) would recycle. This anomaly was experienced during shock applications no. 1 through 4, X axis. This anomaly actually verified that the timing relay was functioning properly.

c. Motor starter (MS-1). During actual shock applications Nos. 7 through 13, Z and Y axes, a momentary discontinuity on motor starter contacts was experienced. This did not affect operation of the control console. There was no functional degradation noted during shock applications in the X axis.

d. Load limiting relay (LLR). There was no functional degradation noted prior to, during, or after actual shock applications. However, due to momentary opening of the pressure electric switch (PE-1) the load limiting relay (LLR) dropped out until the TR-3 timing relay would complete its timing cycle. This occurred during shock applications nos. 1 through 4 of the X axis.

e. Condenser pressure transducer (N6530-1). Up to and including shock applications nos. 1 through 6, Z axis, there was no functional degradation noted. However, upon performing a functional test at the completion of shock No. 7, 100 percent level, Z axis, the output voltage of the pressure transducer (N6530-1) dropped from 1.0 vdc output at 15 psi to 0.701 vdc output at 15 psi. The output pressure versus electrical output was noted to change slightly in the final shock applications, with the final reading, after shock No. 13, as follows: 15 psi = 0.974 vdc. There was no other functional degradation noted.

f. High pressure switch (HDC). During shock No. 10, 75 percent level, Y axis, the high pressure switch (HPC) momentarily opened up as indicated on fault indicator light. This anomaly caused the CR-8 control relay to drop out and the chiller run light to shut off and chiller stop light to go on. The system was reset and testing continued with no other degradation noted to the high pressure switch.

4. References and credits.

Army Corps of Engineers (COE). *Test Report on Safeguard Shock Environmental Testing of 660-ton Chiller Components*, HNDSP-74-308-ED-R. Huntsville, AL: COE, Dec 1974.

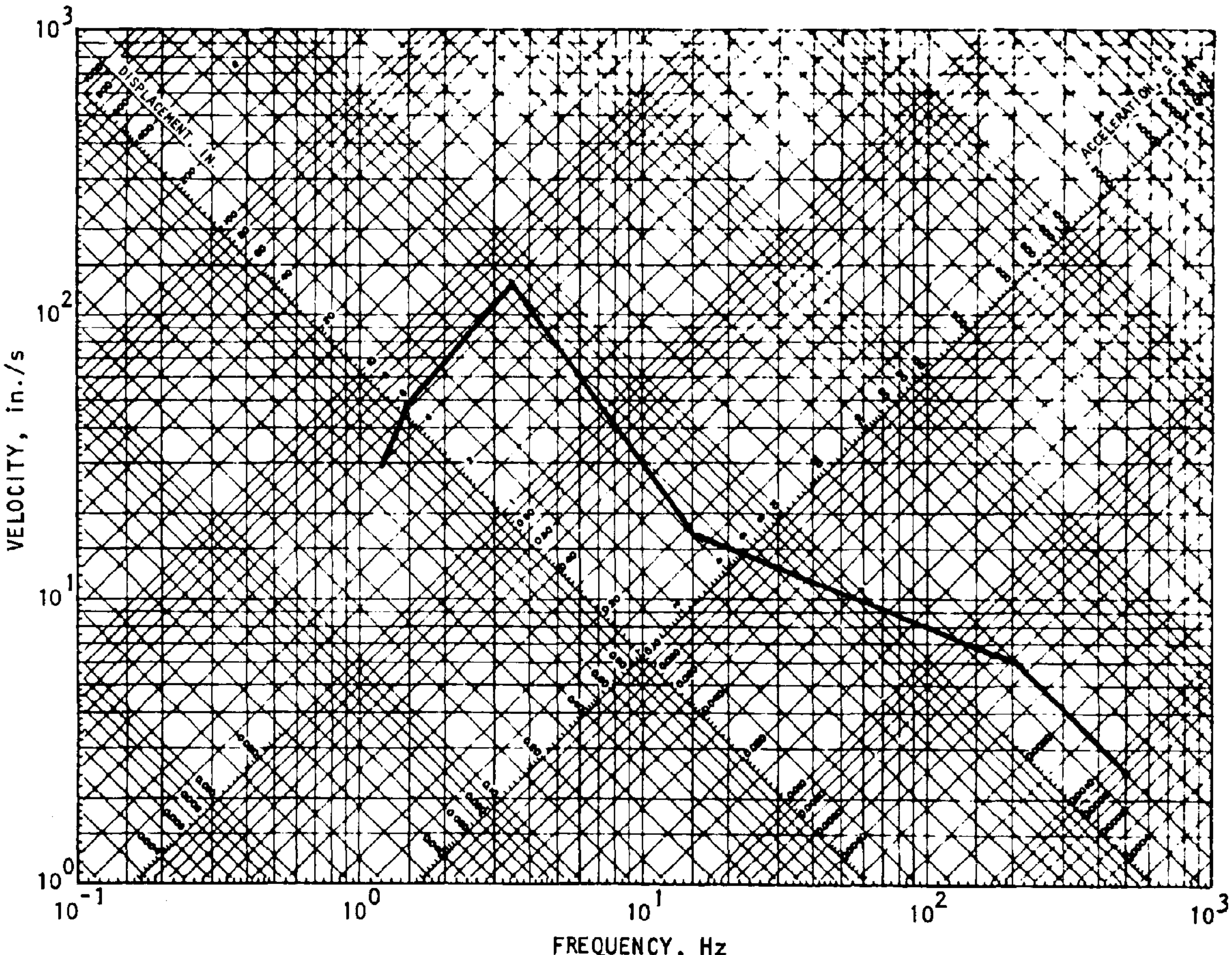

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FIGURE A-3S-1. HORIZONTAL SHOCK SPECTRUM

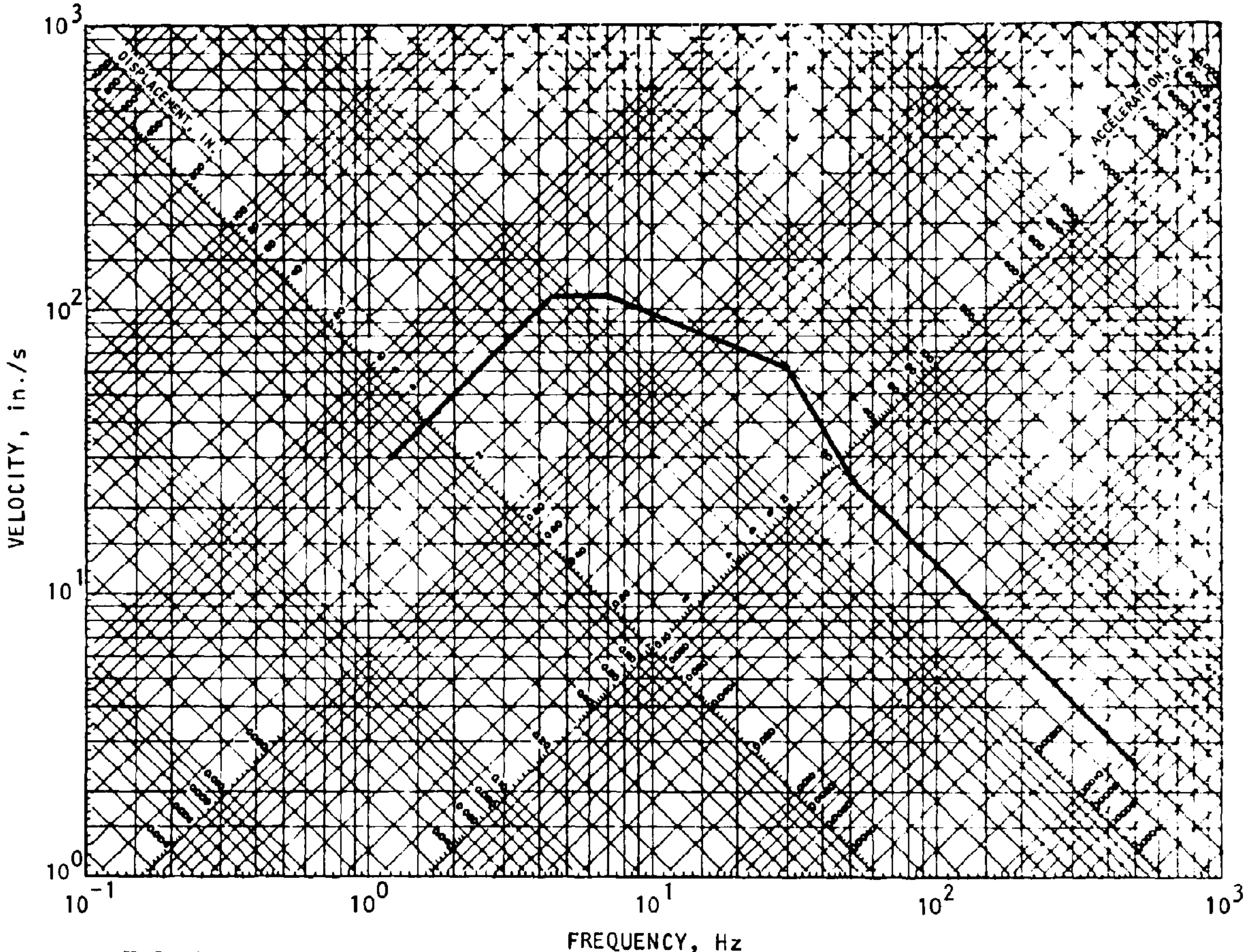

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FIGURE A-3S-2. VERTICAL SHOCK SPECTRUM

A-3T. COOLING COILS

1. AHU cooling coils. This specimen was a Trane, Type W, Series 15, 4-row cooling coil. The outside dimensions are approximately 102 in. wide, 6 in. deep, and 34 in. high. The assembly weighs approximately 300 lb when empty.

2. Test.

a. Shock spectra.

Horizontal	Figure A-3T-1
Vertical	Figure A-3T-2

b. Single axis tests.

3. Results. Passed.

4. References and credits.

The Ralph M. Parsons Co. *NIKE-X Data Report: Shock Tests of Six Selected Items of Equipment*, NX-SE-165. Los Angeles, CA: Parsons, Feb 1969.

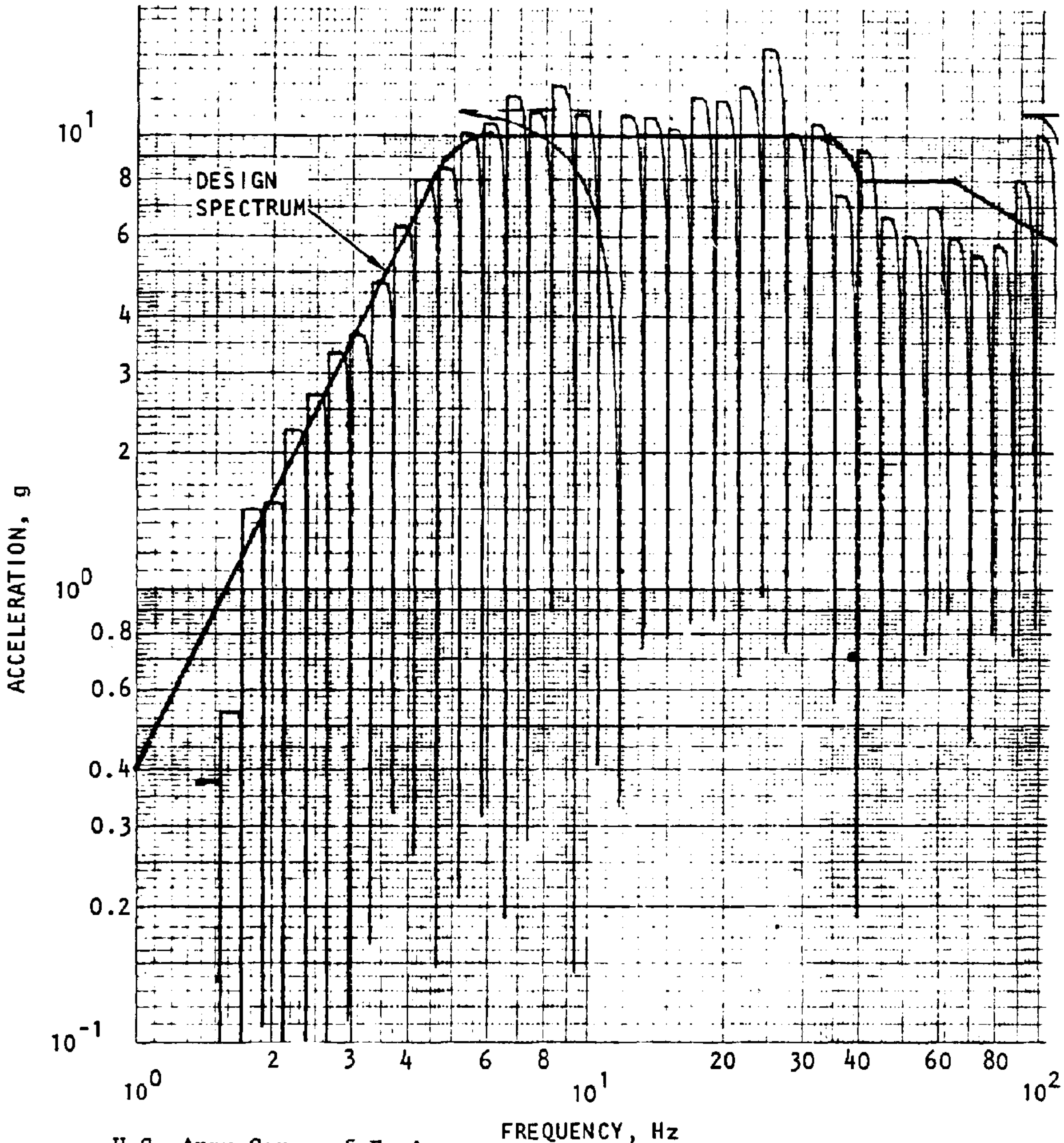

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FIGURE A-3T-1. COMPARISON OF DESIGN AND TEST RESPONSE SPECTRA, HORIZONTAL AXIS

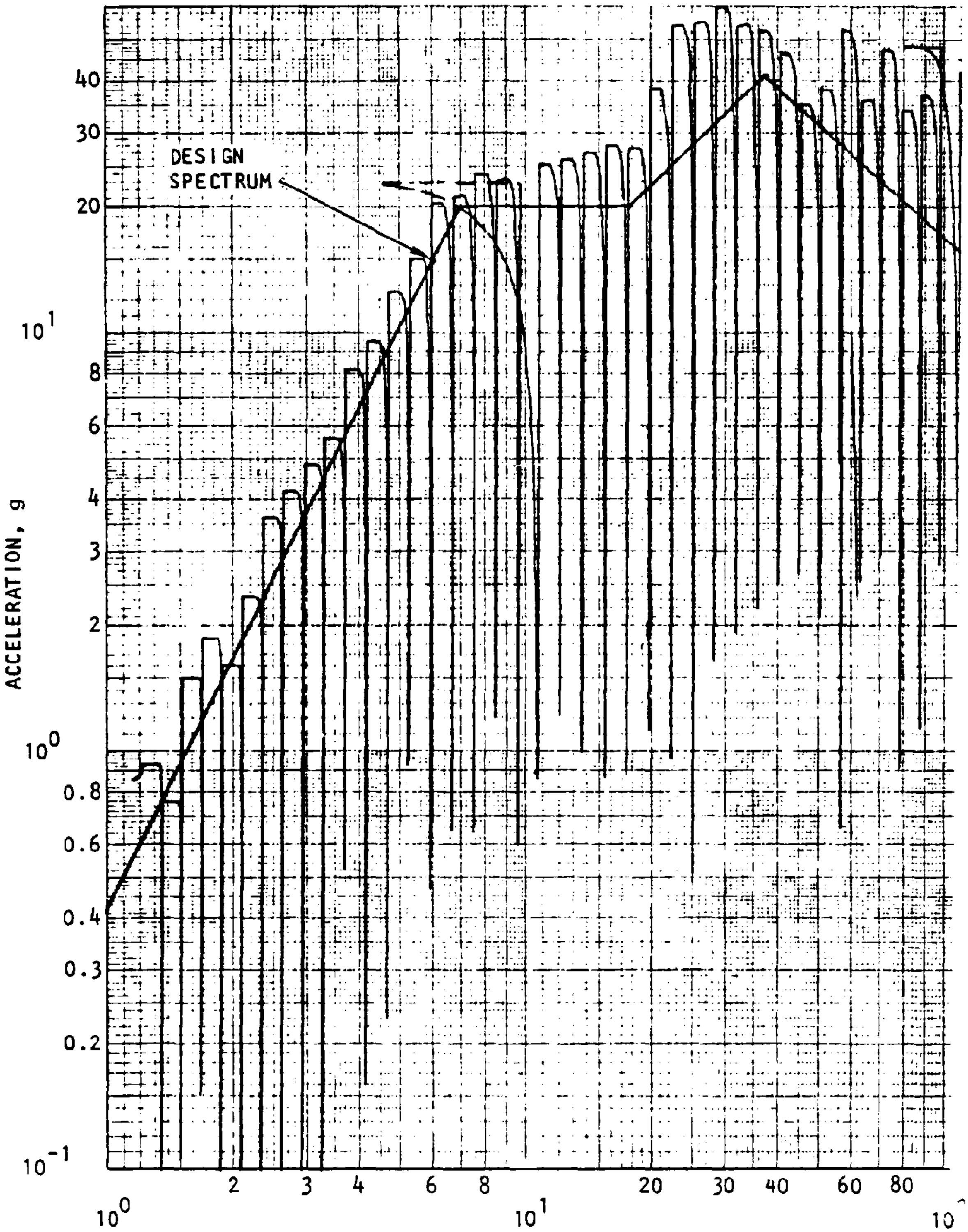

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FIGURE A-3T-2. COMPARISON OF DESIGN AND TEST RESPONSE SPECTRA, VERTICAL AXIS

A-3U. AIR HANDLING UNIT

1. Air handling unit.

a. The air handling unit is designated as Model 39BB1354TA12-M by the manufacturer, the Carrier Air Conditioning Company, a division of the Carrier Corporation, Syracuse, NY. The unit consists of three sections. The first section contains permanent aluminum mesh filters to remove solid particles from the air stream. The second section contains two sets of finned-tube coils for heating and/or cooling the air. The coils were filled with water for the test. The third section contains fans that draw air through the first and second sections and discharge it into the distribution ducts. The assembled unit is 13 ft 10 in. long, 10 ft 4 in. wide, and 6 ft 4 in. high. The unit weighs 10,073 lb with water in the coils.

b. The air handling unit is designed to move 34,000 cu ft of air/min with a static pressure of 2.8 in. of water. It is powered by a 40-hp, 1770-rpm, 460-V, 3-phase, 48-amp, 60-cycle General Electric induction motor, model 5K364AK962, serial No. FH243012. The motor has a continuous time rating and a service factor of 1.15.

c. The drive motor was initially located on top of the air handling unit. It was subsequently relocated on the side of the unit to make the mounting configuration similar to that of the air handling units installed at the Safeguard Grand Forks site.

2. Tests.

a. *Shock spectra*. Figure A-3U-1.

b. *Biaxial tests*.

3. Results.

a. Ten instances of functional or structural failure occurred during the testing of the air handling unit. Before the relocation of the motor, tension in the drive belts was lost twice, at test levels of 75 percent and 100 percent. A leak in the lower coil section occurred at the 75 percent test level. Damage to the filter section occurred twice at the 75 percent test level and 5 times at the 100 percent test level.

b. Loss of tension in the drive belts caused an immediate loss of power to the fan wheel, thus rendering the air handling unit inoperable. The fan drive motor was mounted on the top of the fan section close to the discharge nozzle. Films taken during the testing revealed that the motor tipped along its axis by approximately 5 to 10 deg during the shock; this tilting was apparently due to flexion of the flat top of the fan section. This tilt slackened the belts and disaligned the pulleys sufficiently so that all three belts were dislodged.

c. Development of a coil leak in an actual installation would interfere with normal operation, but would not cause an immediate shutdown of the ventilation equipment. However, while the broken coil is valved off pending repair, the heating or cooling capability of the air unit would be lost.

d. Loss of filters could be potentially more hazardous than a coil leak. During preliminary operation of the fan to check rotation of the motor and electrical wiring, fuses blew after a short time. This problem ended when the filters were installed. Measurement of electric current disclosed that without filters the motor drew considerably higher current because it was moving more air than it was designed to handle. Installation of the filters increased the resistance to flow, thus reducing the quantity of air being moved. Should fuses fail to blow when a filter has disintegrated, overheating of cables or connections could lead to fire hazards. A threaded rod brace was installed in the filter section to pull the filter tracks together, thus holding the filters more securely. Tests indicated that this action had limited success. While it prevented complete loss of filters, partial dislodging of filters from the supporting tracks still occurred.

4. References and credits.

Army Corps of Engineers (COE). *Test Report on Safeguard Shock Environment Testing of Air Handling Unit*, HNDSP-74-325-ED-R. Huntsville, AL: COE, Dec 1974.

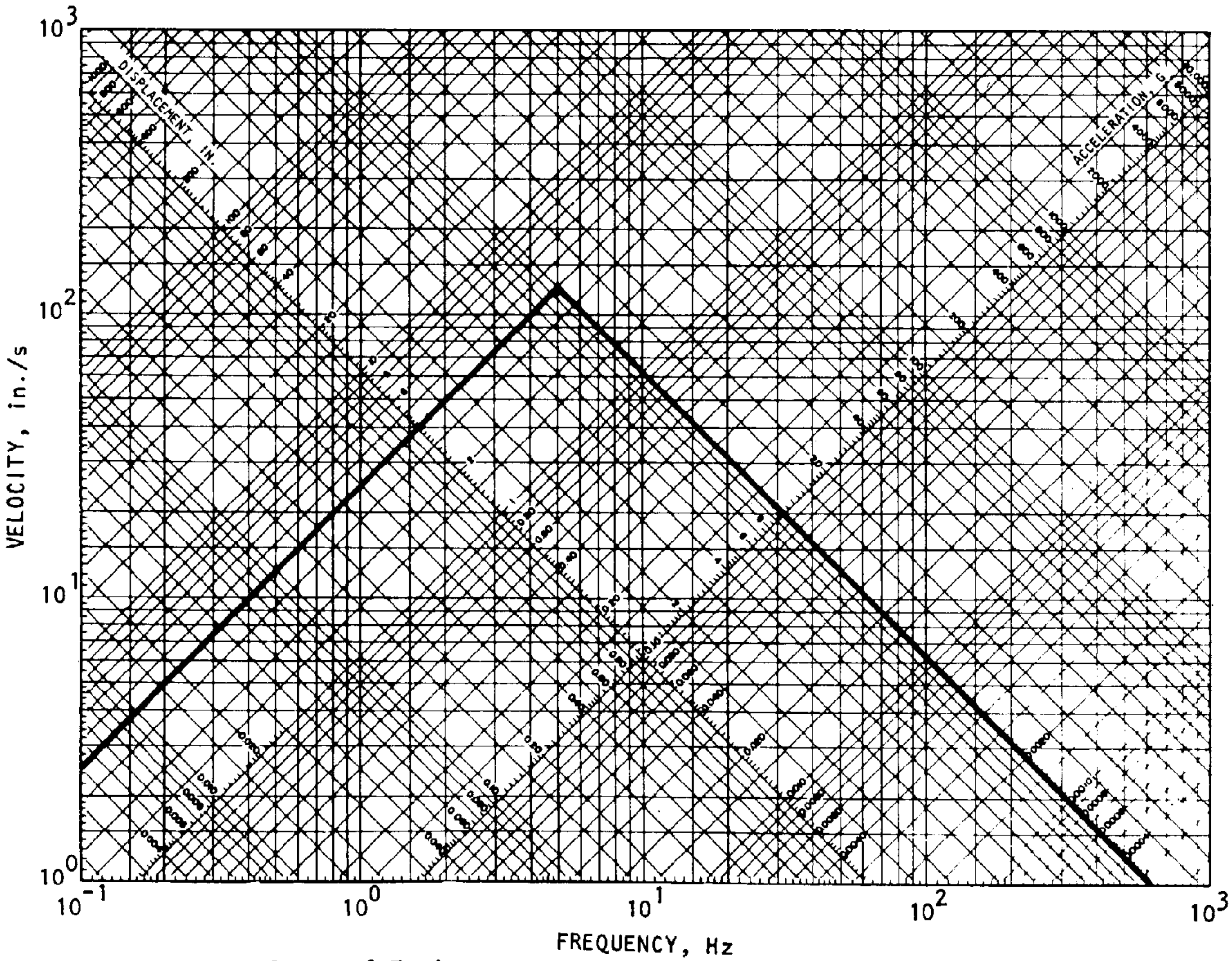

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FIGURE A-3U-1. SHOCK SPECTRUM

A-3V. HEAT SENSOR

1. Heat sensing device assembly. This assembly is described in the table below:

Item Type:	Electric Controller Model 93	Heat Sensor
Space Envelope, in.:	6 by 6 by 18½	6 by 3 by 3
Weight, lb:	16	1
Manufacturer:	Automatic Sprinkler Co.	Automatic Sprinkler Co.

2. Test.

a. Shock spectra.

Horizontal Figure A-3V-1

Vertical Figure A-3V-2

b. Biaxial tests.

3. Results. Prior to and after each shock test the entire assembly was functionally operated to determine that no damage had occurred in previous testing. Also during the actual shock testing both the bellows assembly unit of the heat sensor and the pilot lamp in the electric controller were monitored to determine any malfunction resulting from test conditions. There was no structural degradation noted during or after testing.

4. References and credits.

Army Corps of Engineers (COE). *Test Report on Safeguard Shock Environment Testing of Heat Sensing Device Assembly*, HNDSP-74-345-ED-R. Huntsville, AL: COE, Dec 1974.

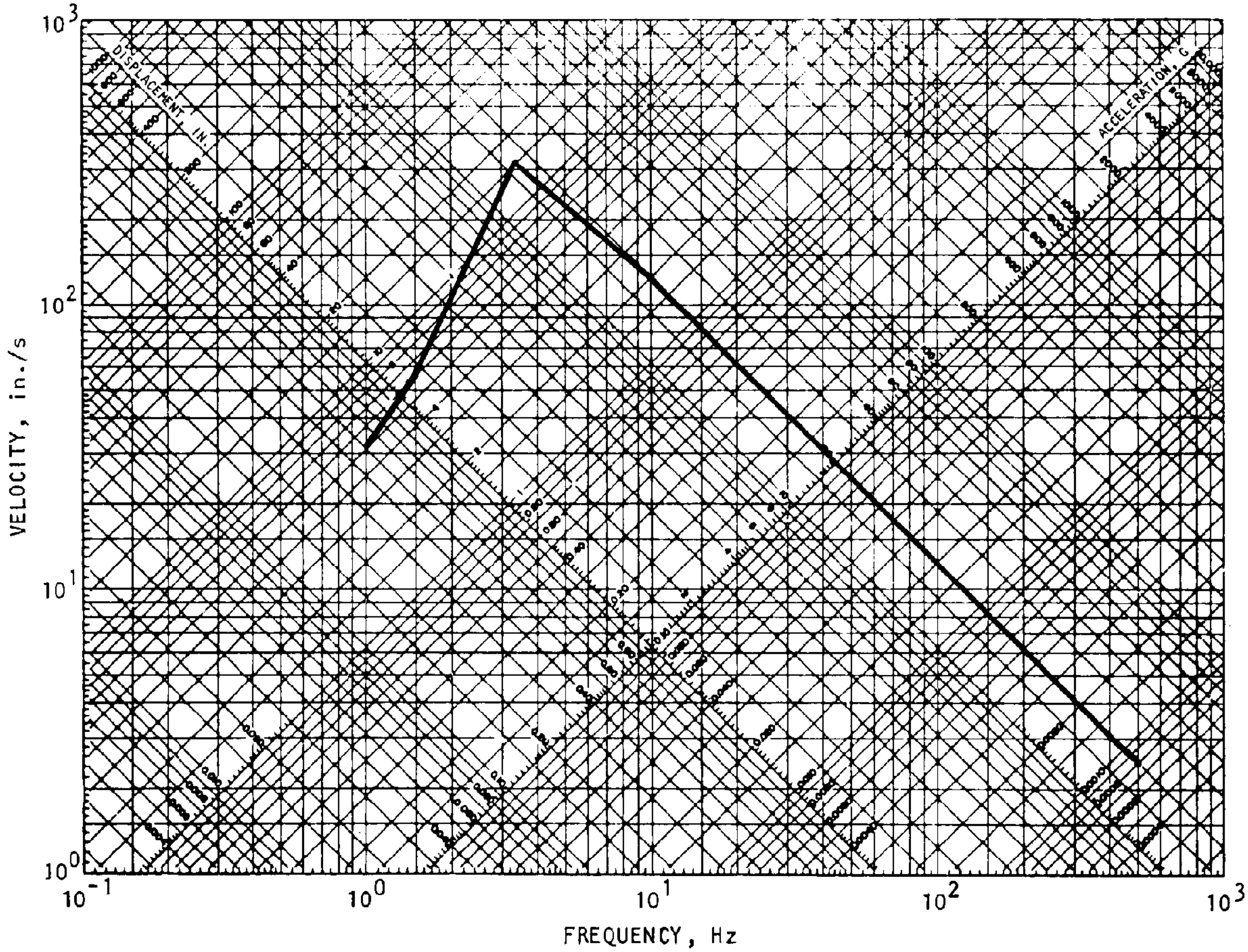

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FIGURE A-3V-1. HORIZONTAL SHOCK SPECTRUM

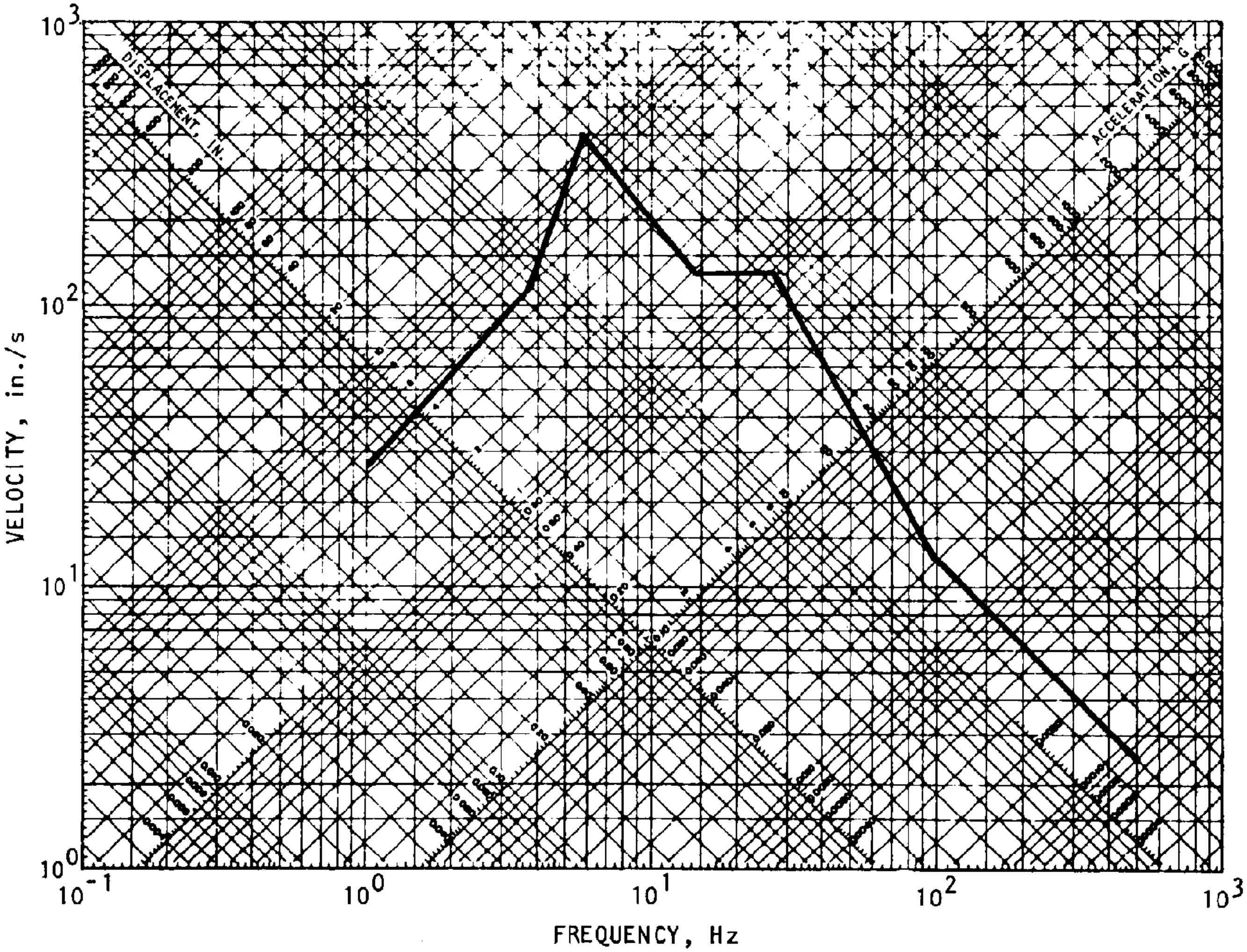

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FIGURE A-3V-2. VERTICAL SHOCK SPECTRUM

A-3W. AIR FILTER

1. AHU air filter. This specimen was a Cambridge Filter Corporation, bag type filter, Model 43X095, with a 24-in. square inlet, a 36-in. long bag, and a 24-in. square, 2-in. thick fiberglass prefilter Model 5K/123232.

2. Test.

a. Shock spectra.

Horizontal	Figure A-3W-1
Vertical	Figure A-3W-2

b. Single axis tests.

3. Results. Passed.

4. References and credits.

The Ralph M. Parsons Co. *NIKE-X Data Report: Shock Tests of Six Selected Items of Equipment*, NX-SE-165. Los Angeles, CA: Parsons, Feb 1969.

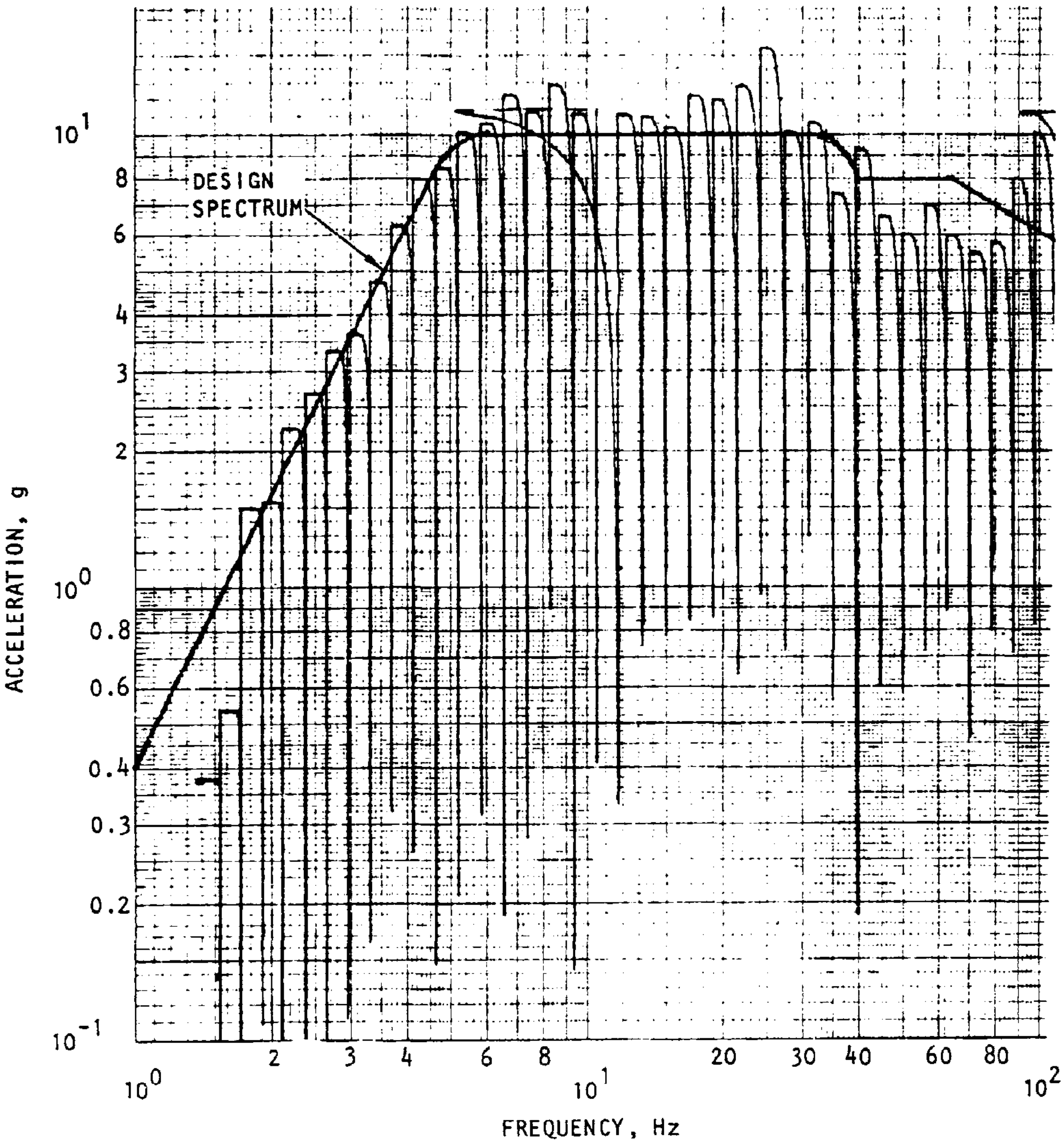

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FIGURE A-3W-1. COMPARISON OF DESIGN AND TEST RESPONSE SPECTRA, HORIZONTAL AXIS

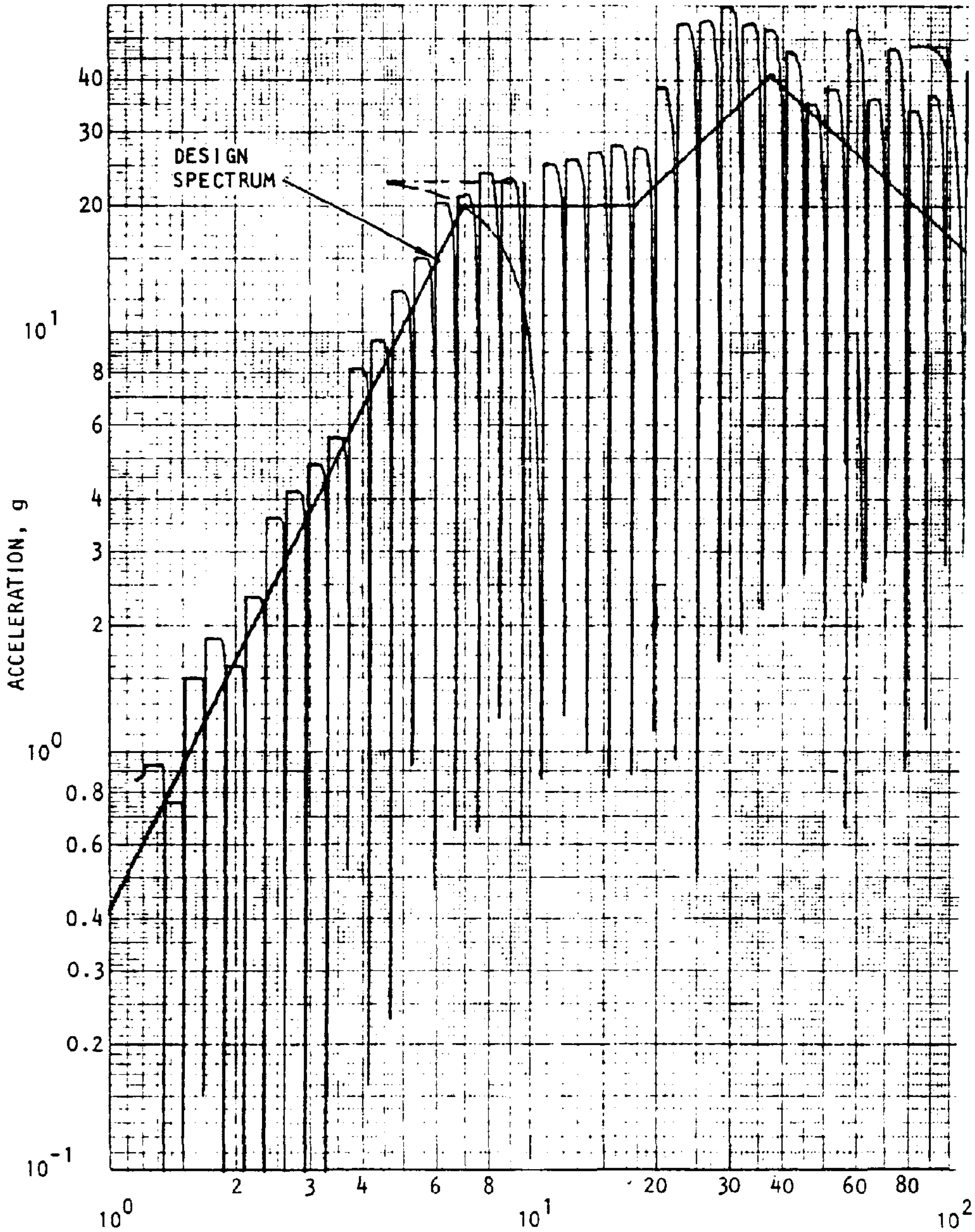

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FIGURE A-3W-2. COMPARISON OF DESIGN AND TEST RESPONSE SPECTRA, VERTICAL AXIS

A-3X. PIPING SEGMENTS

1. Three piping segments.

a. Three separate piping segments were mounted on one structural frame. The principal components of each segment are listed in the table below, with their corresponding Item Type Code (ITC) numbers. The piping segments and supports weight 7220 lb and the frame 6680 lb, making the total weight on the shake table 13,900 lb.

b. The centrifugal pump of the water circulating system is a Pacific Pumping Co. model 20-40955, serial number (S/N) EDG-0637, rated 475 gpm at 43 ft TDH. It is powered by a Reuland electric induction motor, model B3297X, S/N 722359A-1, 7.5 hp, 1800 rpm, 460 V, 3 phase, 60 Hx, 9.7 amp, time rating B.

PIPING SEGMENT PRINCIPAL COMPONENTS

Chiller Water System Segment		*Digital Rack Cooling System Segment*	
Principal Component	ITC No.	Principal Component	ITC No.
Flow indicator	I16FI	Butterfly valve	P05VB
Plug valve	P03VJ	Metal hose assembly	P23HF
Sediment strainer	P06SS	Plug valve	P83VP
Plug valve	P92JV	Pressure indicator	I06PI
Temperature indicator	I02TI	Needle valve	P03VN
Temperature regulating valve	P42CV	Centrifugal pump	P02PC
		Check valve	P87VC
		Pressure indicator	I02PI

Compressed Air System Segment

Principal Component	ITC No.	Principal Component	ITC No.
Gate valve	P69VG	Globe valve	P41VO
Pressure regulating valve	P50VE	Moisture trap	P01TM
		Safety relief valve	P08SR
Needle valve	P13VN	Check valve	P97VC
Pressure indicator	I34PI	Pressure indicator	I35PI
Fluid filter	P15GF	Gaseous storage tank	P21TP
Globe valve	P42VO	Plug valve	P15VV
Fluid filter	P14GF	Sediment strainer	P11SS
Plug valve	P29VJ	Plug valve	P21VJ

2. Test.

a. Shock spectra.

Horizontal	Figure A-3X-1
Vertical	Figure A-3X-2

b. Biaxial tests.

3. Results.

a. Problems experienced.

(1) Both water systems performed satisfactorily, except for a few minor leaks confined to pressure transducer mountings. Also, all of the pressure gages were damaged to some extent during the testing. However, the compressed air system totally failed three times during the testing; in all cases, the failure was in the air receiver and nearby piping.

(2) The first failure occurred during test 11. The "dead-end" air line screwed into the nozzle at the top of the receiver popped out of the nozzle, causing a complete loss of pressure in the system. The two "U" bolts which attached this pipe to the frame were also sheared. Necessary repairs were made and testing continued.

(3) During test 19, the table inserts into which the hold-down bolts for the receiver fixtures were fastened pulled out of the table. This caused the "dead-end" air line to be sheared off at the top nozzle of the air receiver. The threads of this nozzle (a pipe coupling) were damaged too extensively to be repaired quickly. Since a pressure of 15 psig was required during the test, an excessive amount of pipe joint tape was used as a seal, the "dead-end" line was forced into the nozzle, and the compressed air supply from the building was left on for the duration of the test.

b. Discussion of functional and structural anomalies.

(1) Five instances of functional or structural failure occurred during the testing of the piping segments. In each case, the failure occurred at the point where the air line entered the air receiver through the top nozzle. The first time this occurred, the result was a minor leak which was corrected by tightening the pipe. The loss of air from this leak could be overcome by more frequent cycling of an installation's air compressor until the pipe could be tightened.

(2) Complete separation of the pipe from the nozzle would lead to rapid loss of air from the system. Operation of the air compressor could not sustain the pressure, and other systems depending on the compressed air would fail.

(3) However, two important facts must be considered—cumulative damage and overtesting. Twenty-one tests were performed on the test assemblage: three at the 25 percent level, three at the 50 percent level, eight at the 75 percent level, four at the 100 percent level, and three at the 75 percent vertical, 100 percent horizontal level. Damage was noted only at test runs of 75 percent or higher levels. A great amount of overtesting, particularly at the higher frequencies, resulted from limited calibration success.

(4) In addition, the shock level at which failures in the air receiver piping were first noted (75 percent) is considerably higher than the design shock level at the actual location of the air receiver. Considering this fact, no damage to the air receiver piping occurred at its full design shock.

4. References and credits.

Army Corps of Engineers (COE). *Test Report on Safeguard Shock Environment Test of Chilled Water System Segment, Digital Rack Cooling System Segment, and Compressed Air System Segment*, HNDSP-74-326-ED-R. Huntsville, AL: COE, Dec 1974.

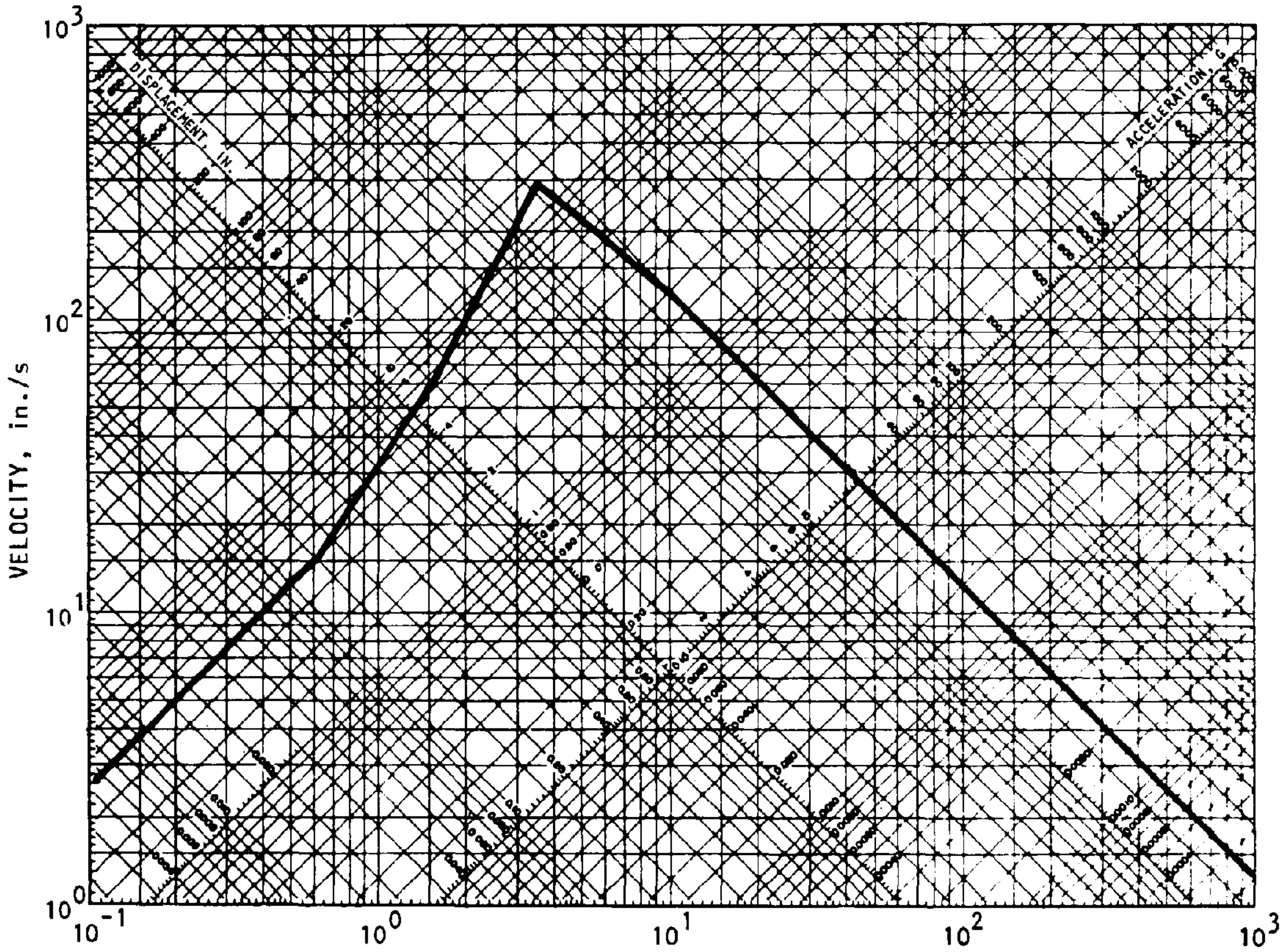

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FIGURE A-3X-1. HORIZONTAL SPECTRUM

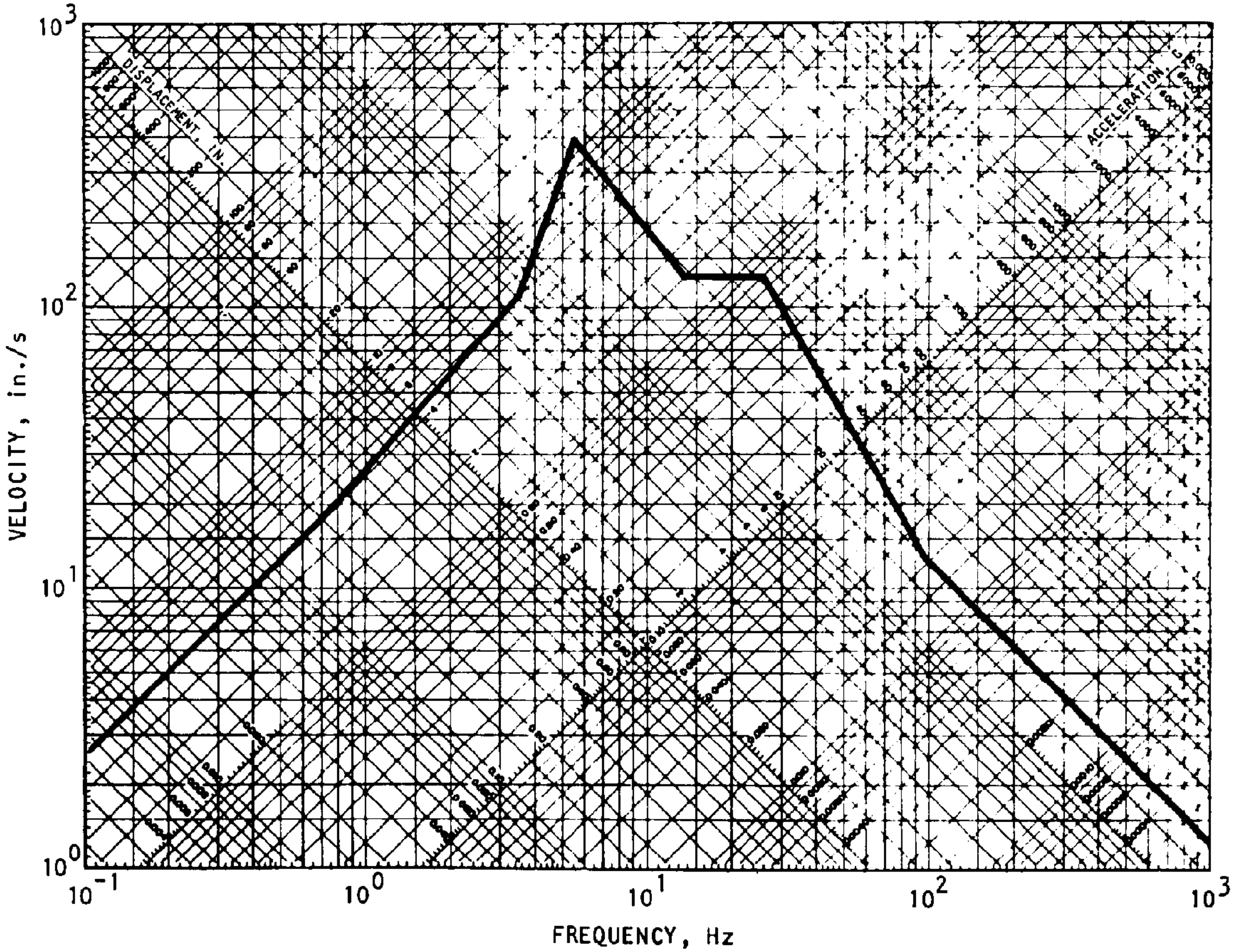

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FIGURE A-3X-2. VERTICAL SPECTRUM

A-3Y. PIPING SEGMENT

1. Piping segment.

a. The chilled water circulating system consists of the components listed in the table below, together with the necessary piping, assembled within a structural steel frame. The system was connected to a water storage tank by means of flexible hoses located at the BSTM shake table periphery; water from the tank was circulated through the system before, during, and after each test. The weight of the system filled with water is 7300 lb, and the weight of the structural frame is 5600 lb, making the total weight on the shake table 12,900 lb. The rate of flow of circulating water was determined using a venturi-type flowmeter and manometer.

b. The pump is Pacific Pumping Company model 29-80122-KPS, serial number (S/N) EDG-0638, rated 1472 gpm at 80 ft TDH. It is powered by a Reuland electric induction motor, model B9870X, S/N 722359C-1, rated 40 hp, 1800 rpm, 460 V, 53 amp, 3 phase, 60 Hz, for continuous service.

MSRPP CHILLED WATER SYSTEM SEGMENT

Principal Component	*ITC No.*
Flow switch	I57FS
Plug valve	P11VJ
Check valve	P76VC
Pressure indicator	I04PI
Needle valve	P13VN
Expansion joint	P56EJ
Centrifugal pump	P39PC
Expansion joint	P57EJ
Sediment strainer	P43SS
Butterfly valve	P07VB

2. Test.

a. Shock spectra.

Horizontal	Figure A-3Y-1
Vertical	Figure A-3Y-2

b. Biaxial testing.

3. Results.

a. Problems experienced.

(1) The most serious problem experienced during the shock testing was shearing of the pump mounting bolt. Shearing of these bolts could lead to misalignment of the pump and motor shafts, causing the coupling to break. This would immediately stop all flow of water through the system, leading to a loss of cooling for the air conditioning system. However, it is suspected that the damage to the motor mounting bolt and flange bolt was due to progressive stretching and loosening of the bolts during the previous tests.

(2) Bending of pipe support "U" bolts and cracking of one or two flange bolts would not have serious consequences or interrupt operation of the system. Loss of a pressure gage would also be of no consequence. If the gage bourdon tube should break, shutting the cock between the gage and the pipe to which it is mounted would stop the loss of water from the line.

b. Deviation from proper performance. Two important facts must be considered—cumulative damage and overtesting. There were 27 tests performed on the test assemblage: six at the 25 percent level five at the 50 percent level, four at the 75 percent level, and 12 at the 100 percent level. Damage was noted only at test runs at 75 percent and 100 percent levels. A great amount of overtesting resulted from limited calibration success, particularly at the higher frequencies.

4. References and credits.

Army Corps of Engineers (COE. *Test Report on Safeguard Shock Environment Test of MSRPP Chilled Water Circulating System*, HNDSP-74-327-ED-R. Huntsville, AL: COE, Dec 1974.

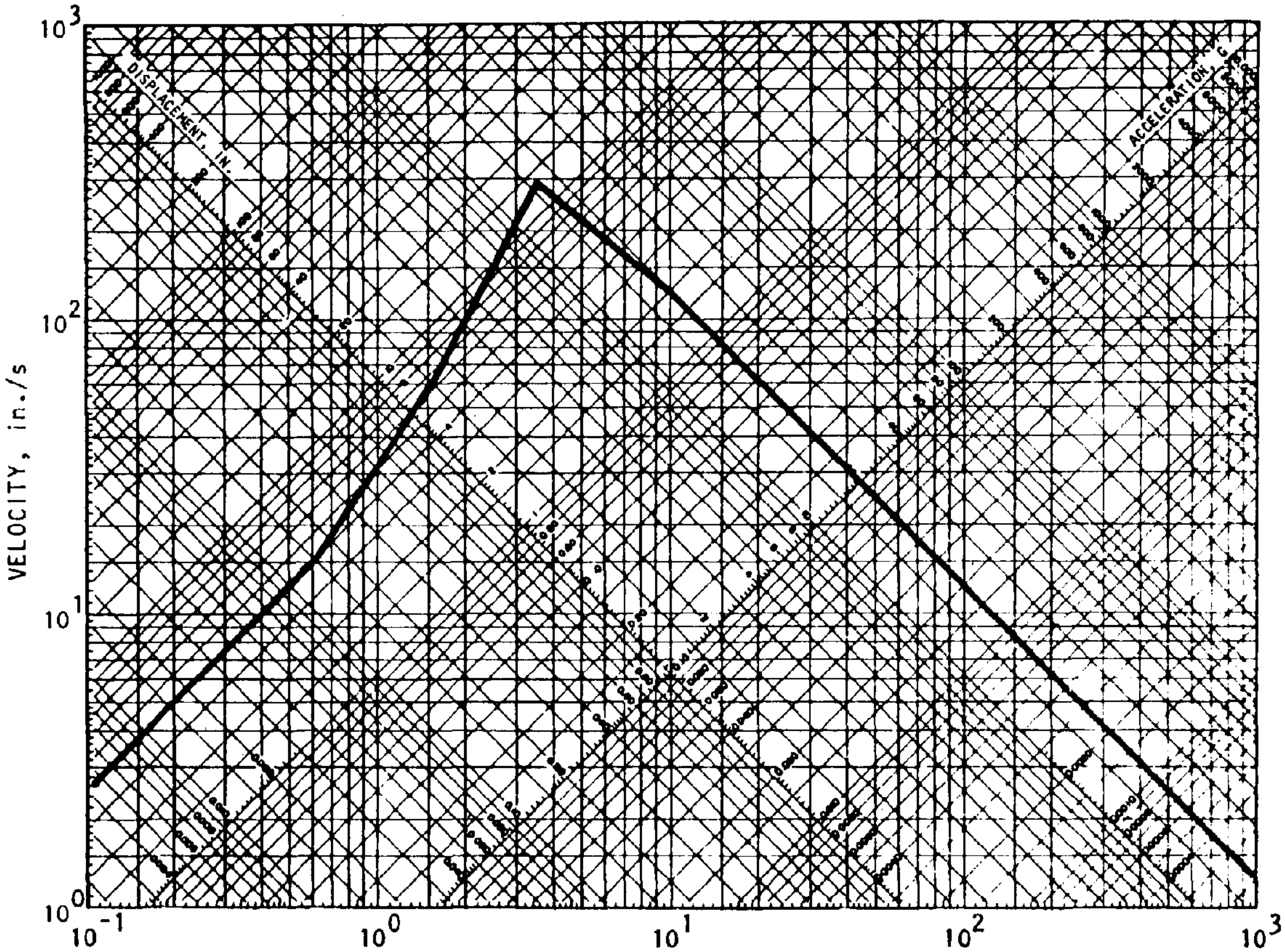

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FIGURE A-3Y-1. HORIZONTAL SPECTRUM

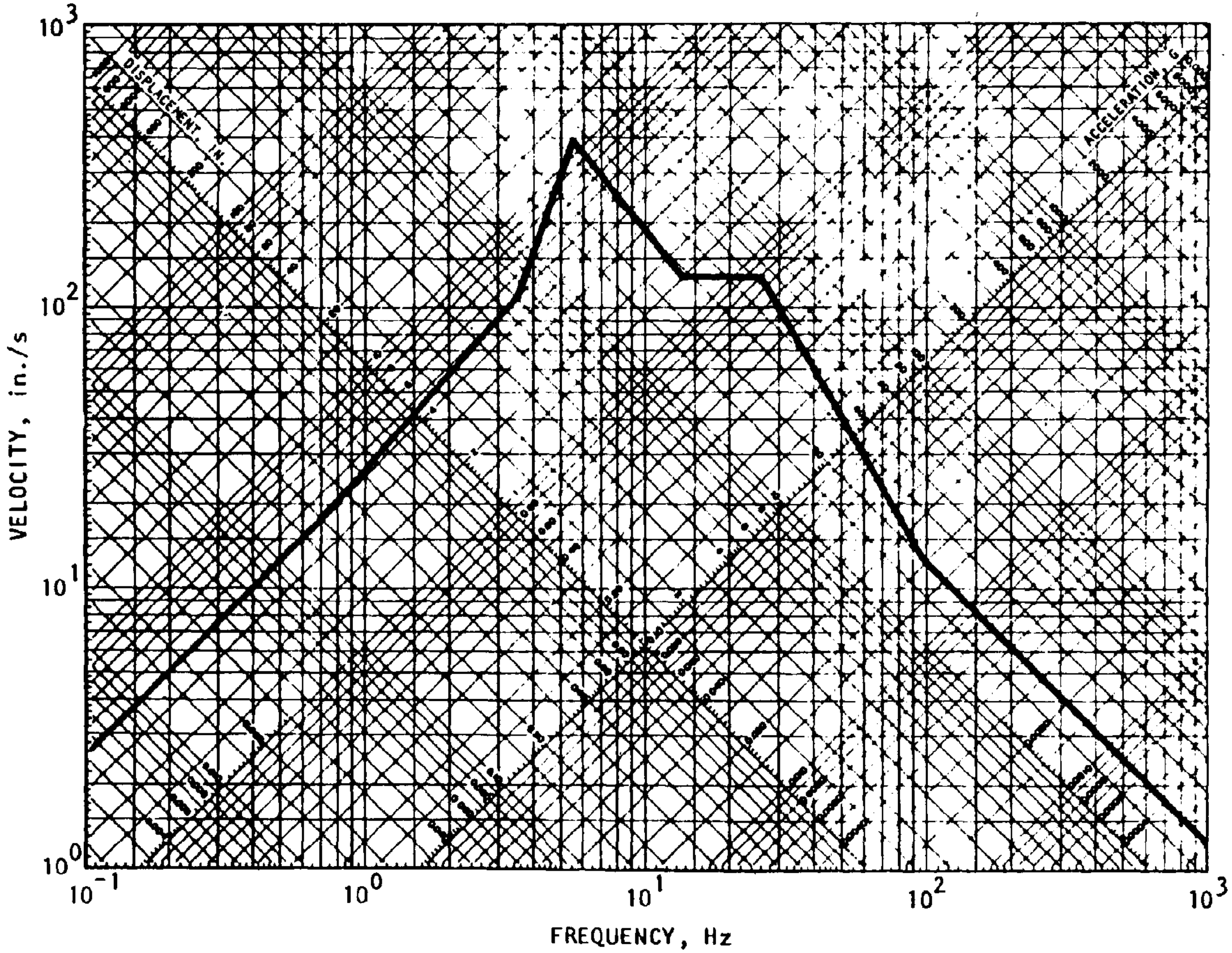

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FIGURE A-3Y-2. VERTICAL SPECTRUM

A-3Z. MONITOR AND CONTROL DUCT-MOUNTED EQUIPMENT

1. Monitor and control duct-mounted equipment. The test specimens consisted of four major test packages which are identified as Test Packages 1, 2, 3, and 4. Specific identification information is shown below.

DESCRIPTION OF TEST SPECIMENS

Item Type	Weight, lb	Space Envelope, in.	Manufacturer
Test Package 1			
I01PE Temperature regulating valve	9.0	14 x 10 x 10	Conoflow Corp.
I04ET Electro-pneumatic transducer	3.0	8 x 4-1/2 x 4-1/2	Ametek
I07TT Temperature transducer	12.0	14 x 14 x 8	Ametek
I31TT Temperature transducer	12.0	14 x 14 x 8	Ametek
I13TT Temperature transducer	35.0	16 x 12 x 10	Honeywell
I01TE Temperature element	2.0	10-1/2 x 10-1/2 x 3	Honeywell
I14PT Pressure transducer	4.0	9 x 7 x 8-1/2	Robertshaw
Extractor (EX-88A)	7.0	18 x 1-1/2 x 18	Krueger
Volume damper (VC-8)	2.0	22 x 4 x 1-1/2	Krueger
Four-way diffuser (S-E4)	2.0	6 x 4 x 6	Krueger
Four-way diffuser (S-E4)	8.0	18 x 4 x 18	Krueger

A-3Z (Cont'd) MONITOR AND CONTROL DUCT-MOUNTED Q EQUIPMENT

Item Type	Weight, lb	Space Envelope, in.	Manufacturer
Test Package 2			
I50AP Pressure switch	3.0	7 x 3 x 7	Mercoid
I01MT Humidity transmitter	2.8	12-1/2 x 9 x 11-1/2	Powers
I02CH Dew point sensor	2.0	5-1/2 x 3 x 4-1/2	Telmar
I02CH Dew point sensor probe	0.8	14 x 4 x 4	Foxboro
I10CH Relay and power adapter	2.0	8.5 x 8.5 x 3	Barber Colman
I10CH Humidity transmitter	1.8	16-1/2 x 4 x 4-1/2	Barber Colman
I10CH Power supply	4.0	7 x 4 x 3-1/2	Barber Colman
I10CH Receiver-controller	1.0	10-1/2 x 4 x 3	Barber Colman
I23PL Pneumatic thermostat	5.0	13 x 8-1/2 x 10	Fairchild Hiller
P68VE Regulator filter	4.0	9-1/2 x 3-1/2 x 3	Fairchild Hiller
Manual pressure reducing valve	40.0	14 x 16 x 7	Krueger
Manual pressure reducing valve	60.0	34 x 6 x 30	Krueger
Test Package 3			
Top register (880V)	6.0	8 x 5 x 6	Krueger
Top register (880V)	95.0	48 x 3-1/2 x 36	Krueger
Fire damper (119-V-Type A)	3.0	24 x 12 x 12	Air Balance
Volume damper (VC-8)	8.0	30 x 1-1/2 x 18	Krueger
Test Package 4			
I33SL Pressure switch	3.0	5-1/4 x 6 x 4-1/2	Square "D"
I63FS Flow switch	4.0	7 x 7 x 33	Peeco
I20SL Current trip	2.8	4-1/2 x 3 x 5-1/2	Telmar
I01DA Pneumatic pressure regulator	10.0	20-1/2 x 11-1/2 x 9-1/2	Fairchild Hiller
H87DT Damper	70.0	30 x 4 x 28	Imperial Damper

2. Test.

a. Shock spectra.

Horizontal	Figure A-3Z-1
Vertical	Figure A-3Z-2

b. Single axis tests.

3. Results.

a. Temperature transducer I07TT experienced structural damage. The male end of the ¼-in. elbow, that joins the pressure regulator and the transducer housing, sheared off. A bracket was added to support the air regulator and the specimen retested with no recurrence of the trouble.

b. Humidity transmitter I01MT functionally failed during shock in the Y axis. The failure was caused by the sensing element being severed.

c. The straps restraining the release mechanism on the 24 by 12-in. fire damper (119-V type A) deformed during shock testing and slipped out of its slots (in the fusable link) and closed the fire door. The restraint straps and fusable link in the damper were reassembled with a sprial twist applied to the straps. The specimen was then retested with no further functional degradation.

d. During testing of the 48 by 36-in. top register (880-V) and 18 by 18-in. four-way diffuser (S-E4), the set position of the vanes and the blades were displaced. The lower row of the adjustable blades on the register moved from full open to full closed position. The vanes on the diffuser moved from full open to 60 percent closed position. This movement was caused by excessive mechanical slack at the point where the vanes are attached to the linkage.

e. No functional degradation nor structural damage occurred on the remaining components covered by this test.

4. References and credits.

Army Corps of Engineers (COE). *Test Report on Safeguard Shock Environmental Testing of Monitor and Control Components and Duct Mounted Equipment*, HNDSP-74-307-ED-R. Huntsville, AL: COE, Apr 1974.

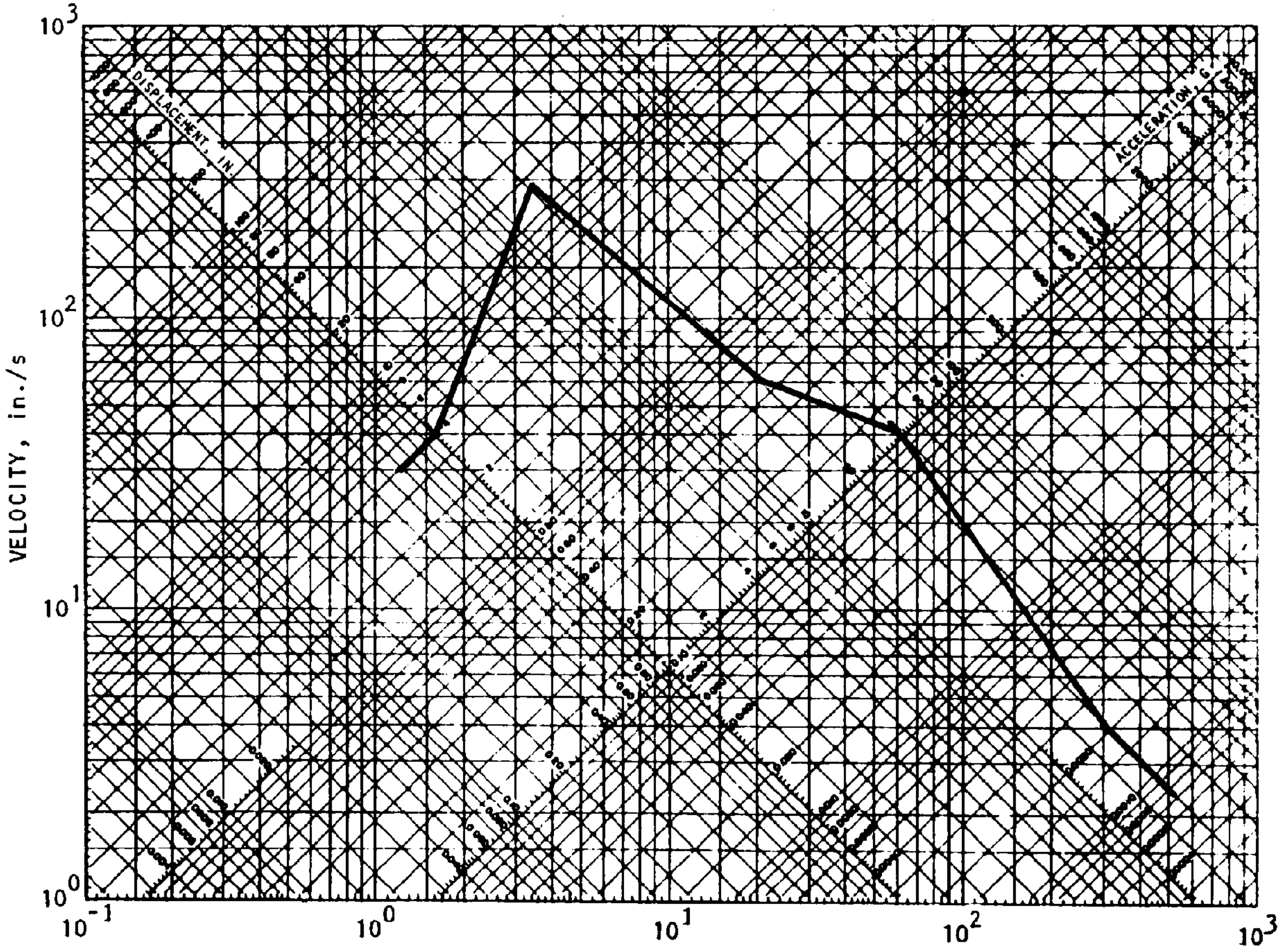

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FIGURE A-3Z-1. HORIZONTAL SHOCK SPECTRUM FOR OUTSIDE AIR MAKE-UP, LOCAL INSTRUMENTS, PRESSURE SWITCHES, AND PNEUMATIC CONTROL VALVE

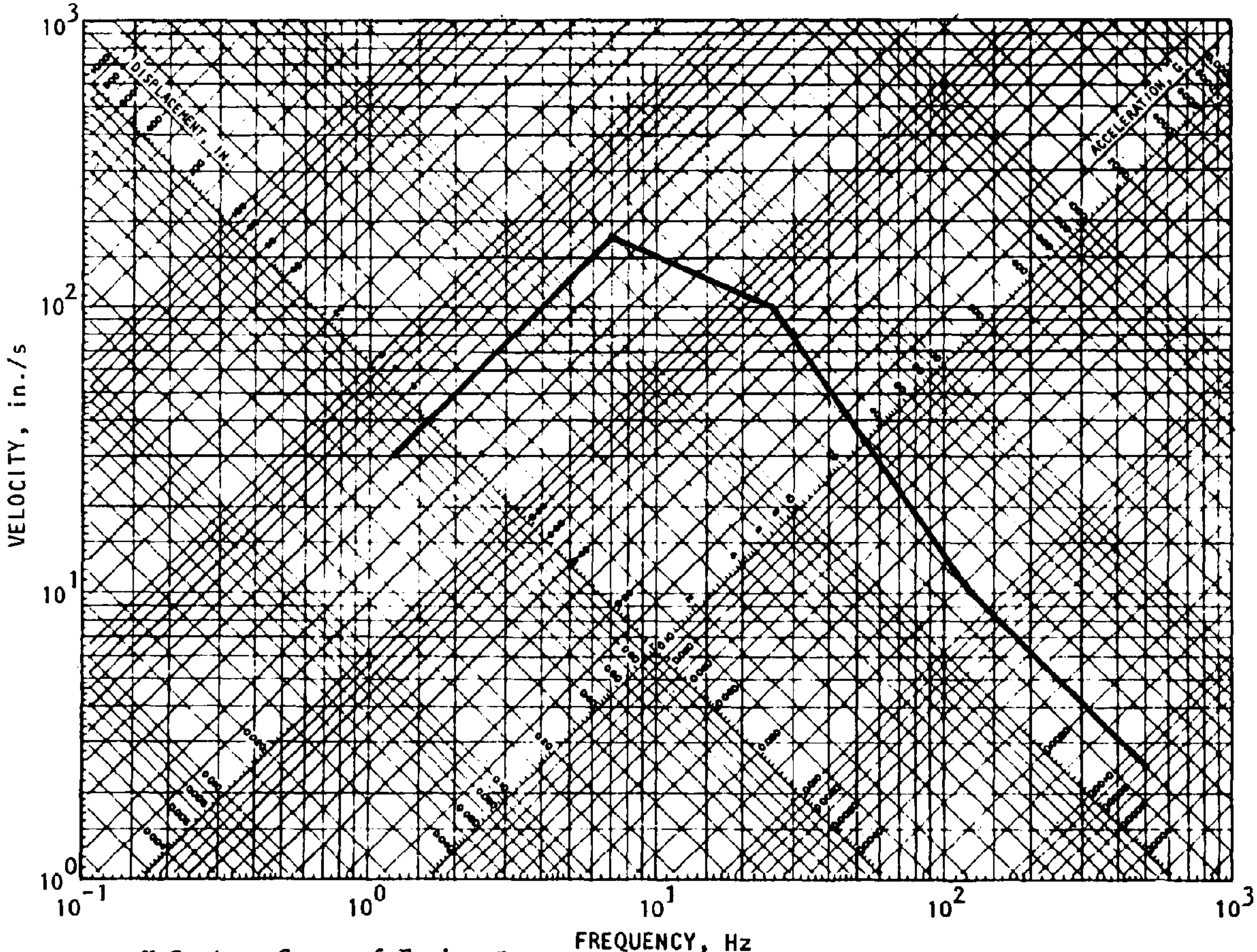

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FIGURE A-3Z-2. VERTICAL SHOCK SPECTRUM FOR OUTSIDE AIR MAKE-UP, LOCAL INSTRUMENTS, PRESSURE SWITCHES, PNEUMATIC CONTROL VALVE, PNEUMATIC ACTUATOR, AND TEMPERATURE TRANSMITTER

A-3AA. PUMP, MOTOR AND VALVE

1. Pump and motor with flow control valve. This combination specimen consisted of a pump, Pacific Pump Company, Model 11-305-533261, 380 gpm at 24-ft TDH, with 5 hp motor, valve; and a Johnson Controls valve with actuator assembly, V5820-15, 3-in. size. The motor-pump assembly was suitably piped to the Johnson flow control valve.

2. Test.

a. Shock spectra.

Horizontal — Figure A-3AA-1

Vertical — Figure A-3AA-2

b. Single axis tests.

3. Results. Passed vertical and one horizontal test. In second horizontal axis, pressure fluctuation about 12 psig reference ranged from 8.5 to 16 psig. System returned to normal after test.

4. References and credits.

The Ralph M. Parsons Co. *NIKE-X Data Report: Shock Tests of Six Selected Items of Equipment*, NX-SE-165. Los Angeles, CA: Parsons, Feb 1969.

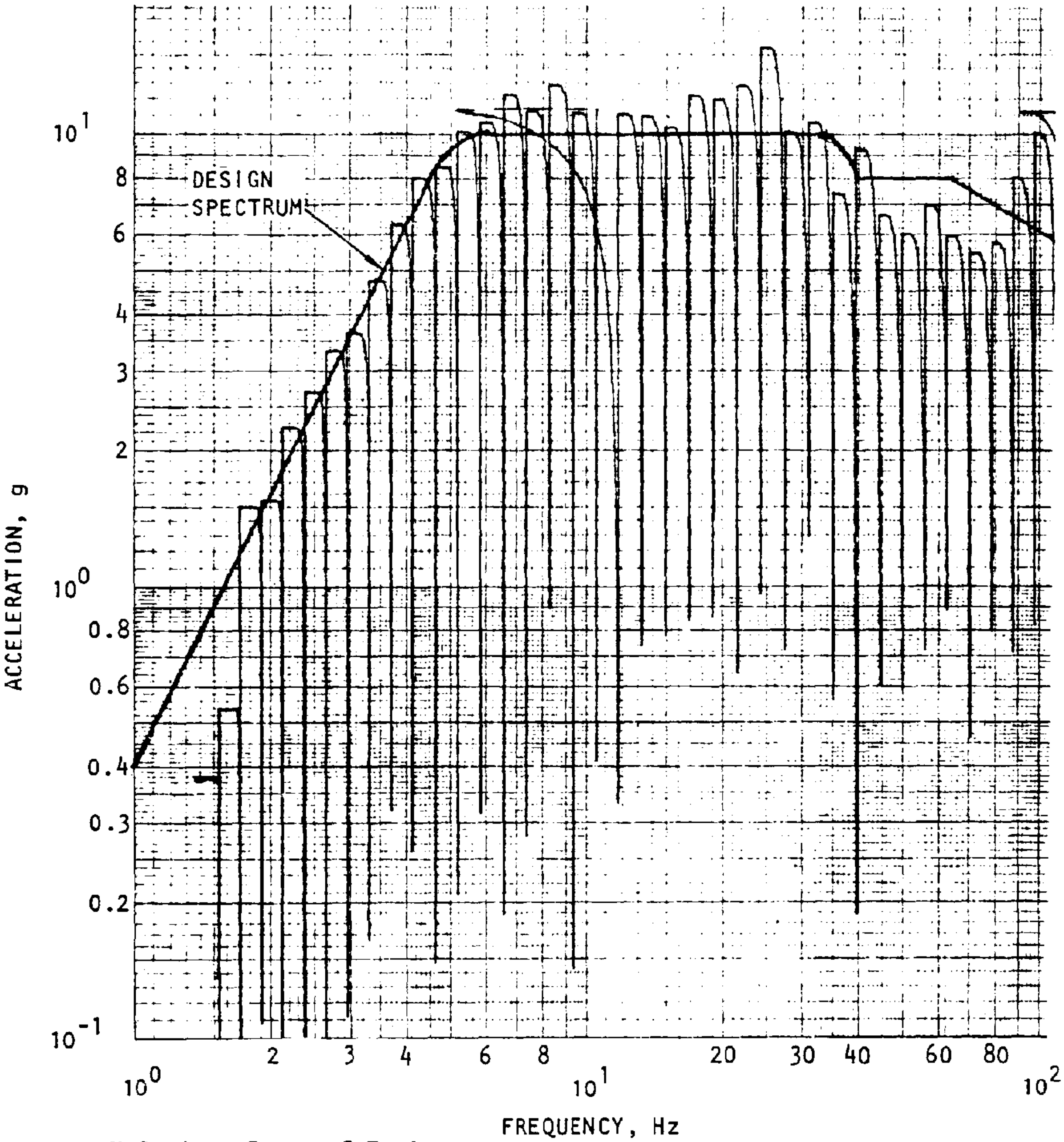

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FIGURE A-3AA-1. COMPARISON OF DESIGN AND TEST RESPONSE SPECTRA, HORIZONTAL AXIS

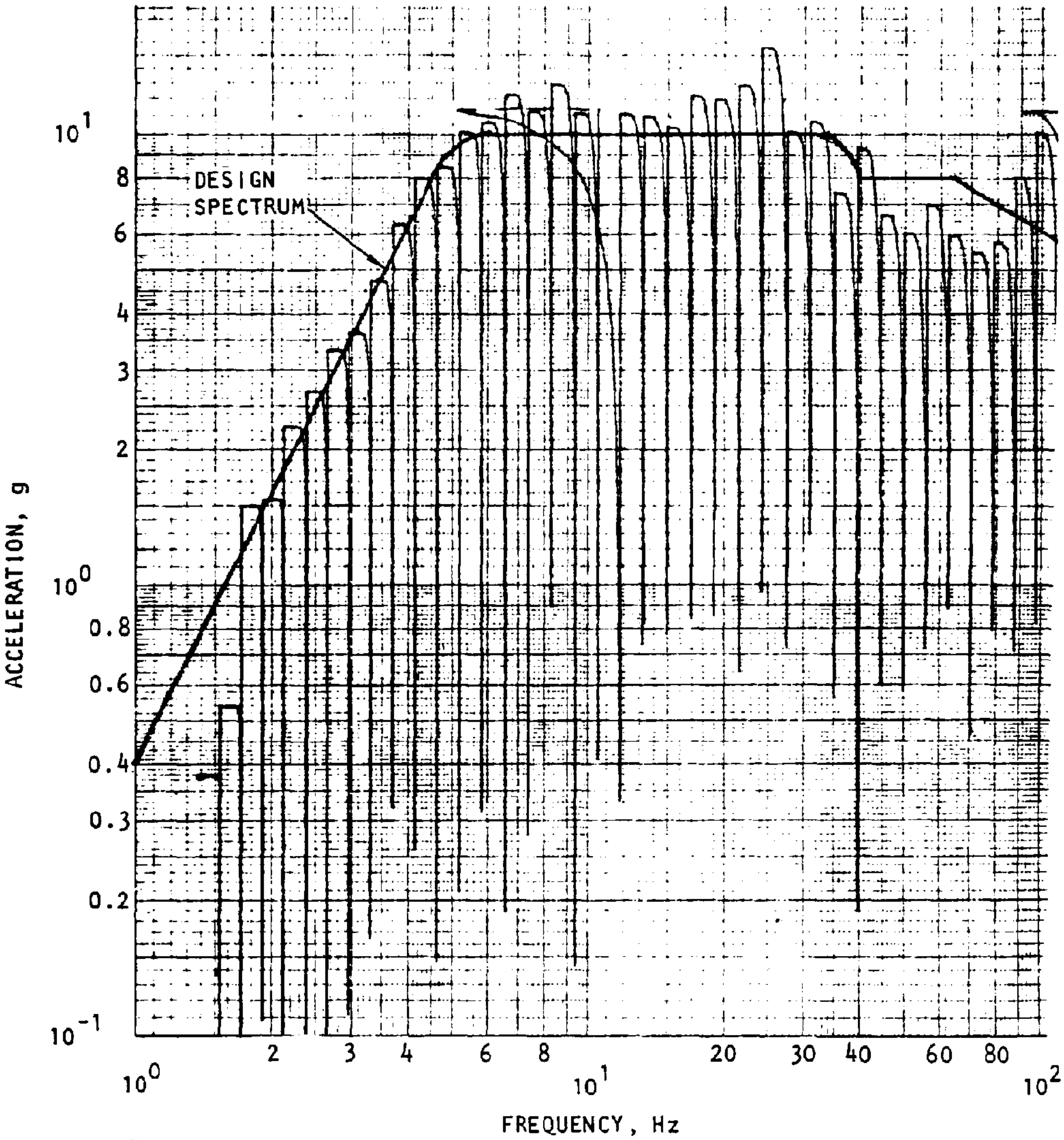

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FIGURE A-3AA-2. COMPARISON OF DESIGN AND TEST RESPONSE SPECTRA, HORIZONTAL AXIS

A-3BB. SHUTDOWN SWITCH

1. Compressor control shutdown switch.

The switch is described in the following table:

Item Type:	Compressor control oil shutdown switch
Space Envelope, in.:	9 by 8 by 11
Weight, lb:	10
Manufacturer:	Chicago Pneumatic

2. Test.

a. Shock spectra.

Horizontal	Figure A-3BB-1
Vertical	Figure A-3BB-2

b. Biaxial tests.

3. Results.

a. Prior to and after each shock test the test specimen and reservoir was filled with oil and then drained to determine that the switch functioned properly. Prior to, during, and after each full level shock application, the test specimen was monitored for any switch discontinuity. There was no structural degradation noted during or after any test.

b. During shock application No. 5, 100 percent, X-Y axis, and shock No. 7, 100 percent, Z-Y axis, a discontinuity was experienced on switch. Switch contacts opened.

c. This anomaly occurred when oil level was just above switch actuation point (closed).

4. References and credits.

Army Corps of Engineers (COE). *Test Report on Safeguard Shock Environmental Testing of Compressor Control Oil Shutdown Switch*, HNDSP-74-340-ED-R. Huntsville, AL: COE, Nov 1974.

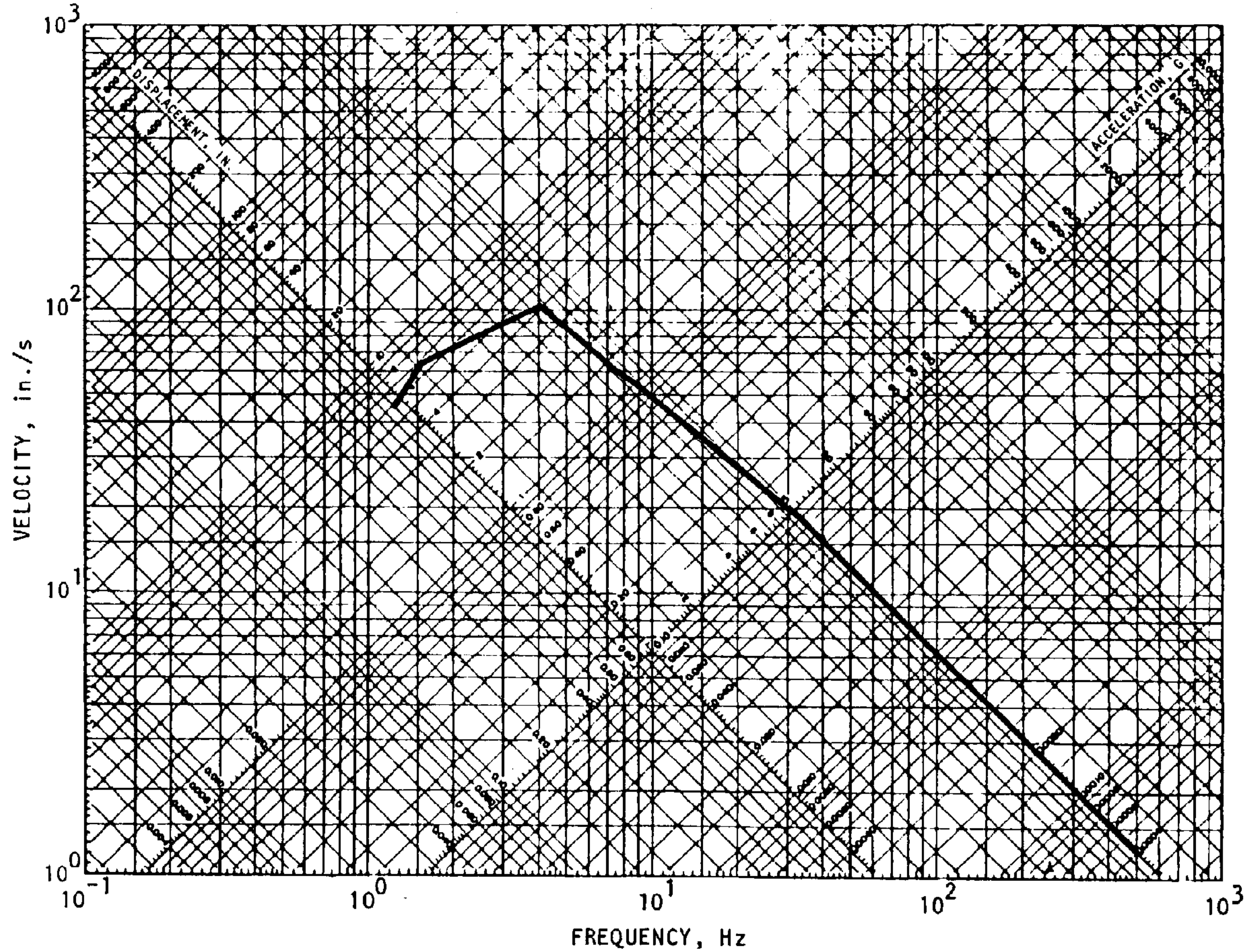

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FIGURE A-3BB-1. HORIZONTAL SHOCK SPECTRA

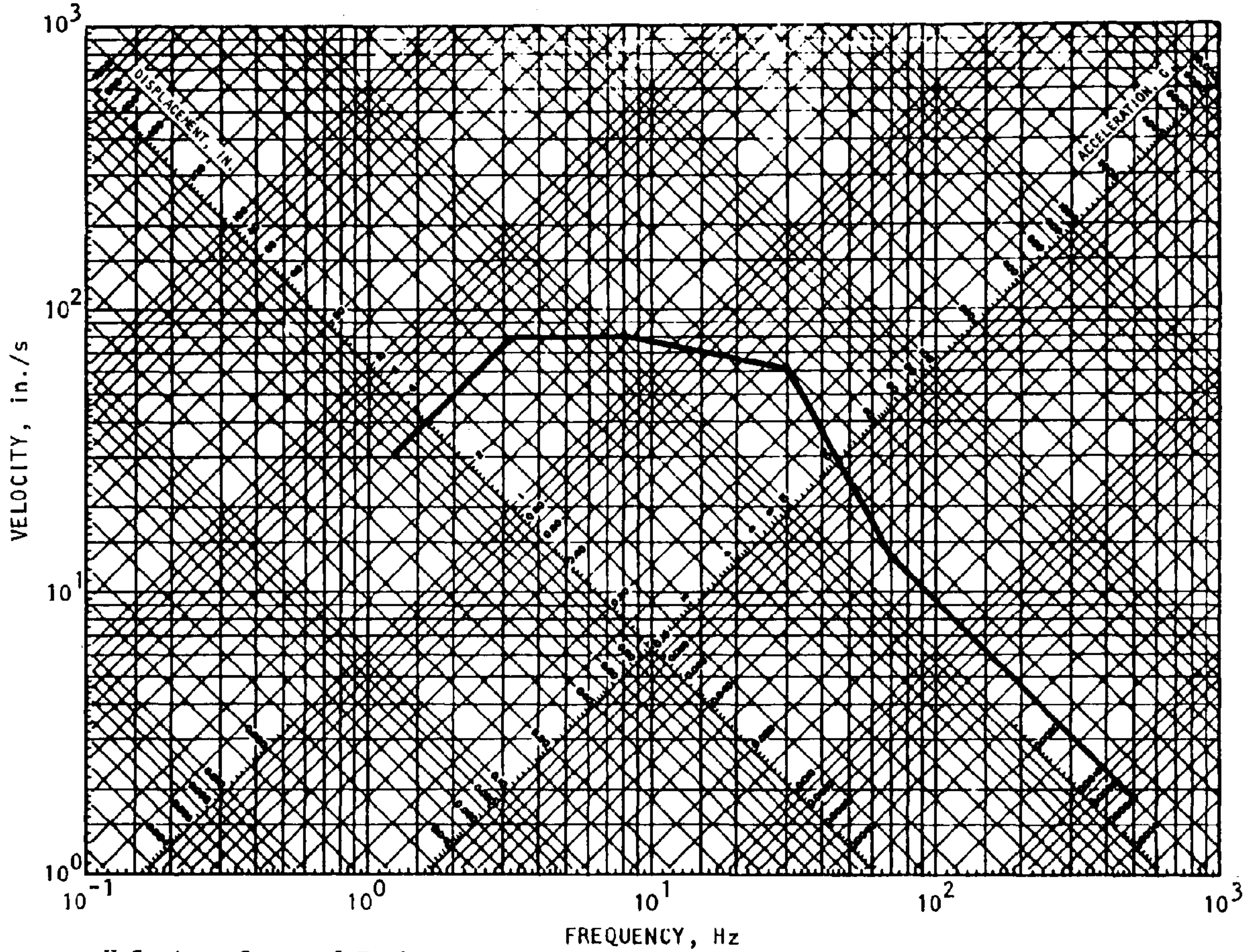

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FIGURE A-3BB-2. VERTICAL SHOCK SPECTRA

A-3CC. PRESSURE CONTROL VALVE

1. Pressure control valve. The valve is described in the following table:

Item Type:	Pressure control valve
Space envelope in:	18½ by 12½ by 24
Weight, lb:	300
Manufacturer:	Jordan Valve Co.

2. Test.

a. Shock spectra.

Horizontal	Figure A-3CC-1
Vertical	Figure A-3CC-2

b. Single axis tests.

3. Results. Prior to and after each shock test the pressure control valve P83VE was functionally operated by flowing water at 150 psig pressure to the inlet and adjusting the valve to regulate the pressure down to 100 psig on the outlet side. Prior to, during, and after each shock application the inlet and outlet static pressures were monitored and transient pressures were monitored during shock applications. There was no structural degradation noted during or after testing. The valve exhibited some deviations from the regulated pressure output, but these were not enough to effect the functional performance of the valve.

4. References and credits.

Army Corps of Engineers (COE). *Test Report on Safeguard Shock Environmental Testing of Pressure Control Valve (P83VE)*, HNDSP-74-342-ED-R. Huntsville, AL: COE, Nov 1974.

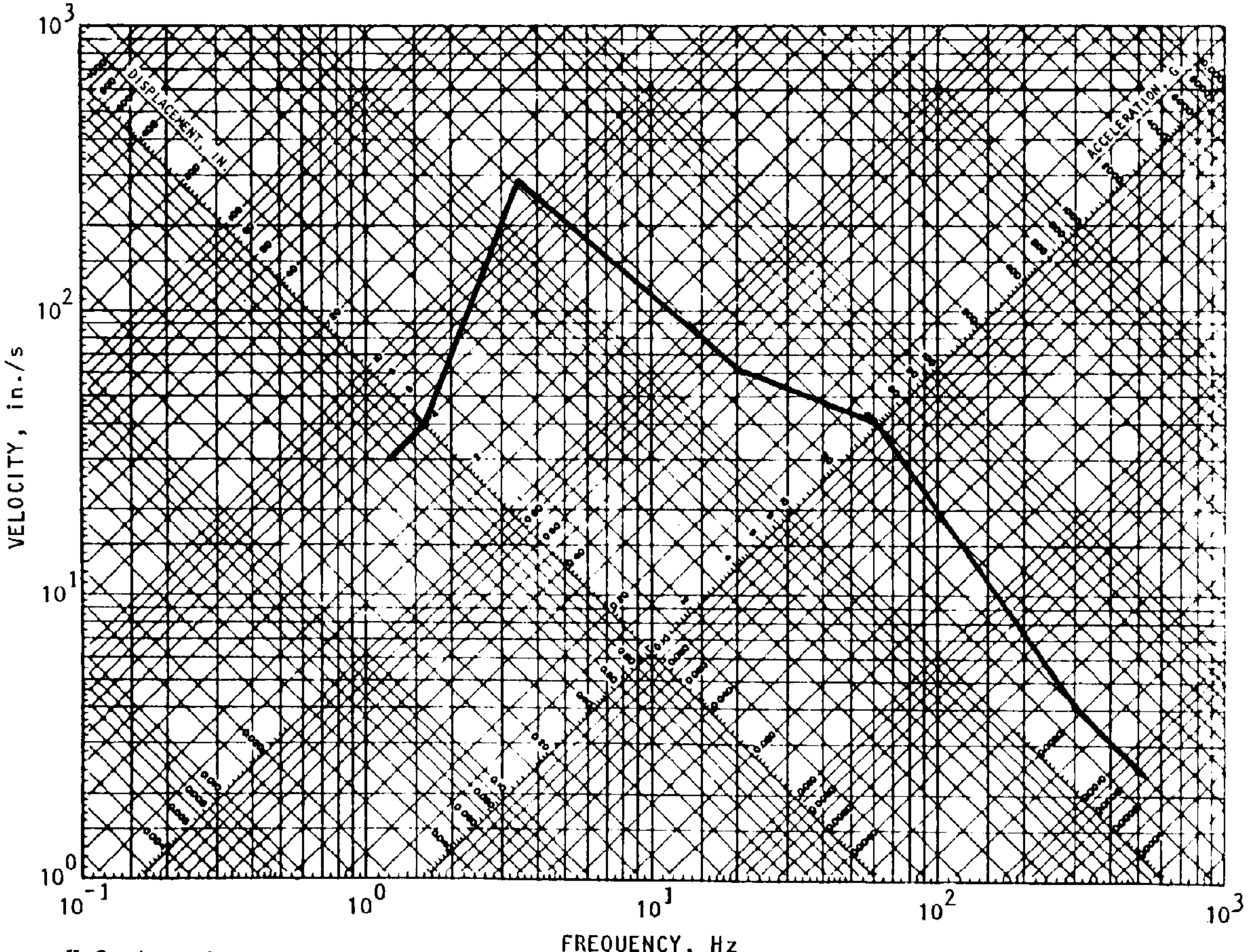

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FIGURE A-3CC-1. HORIZONTAL SHOCK SPECTRUM FOR OUTSIDE AIR MAKE-UP, LOCAL INSTRUMENTS, PRESSURE SWITCHES, AND PNEUMATIC CONTROL VALVE

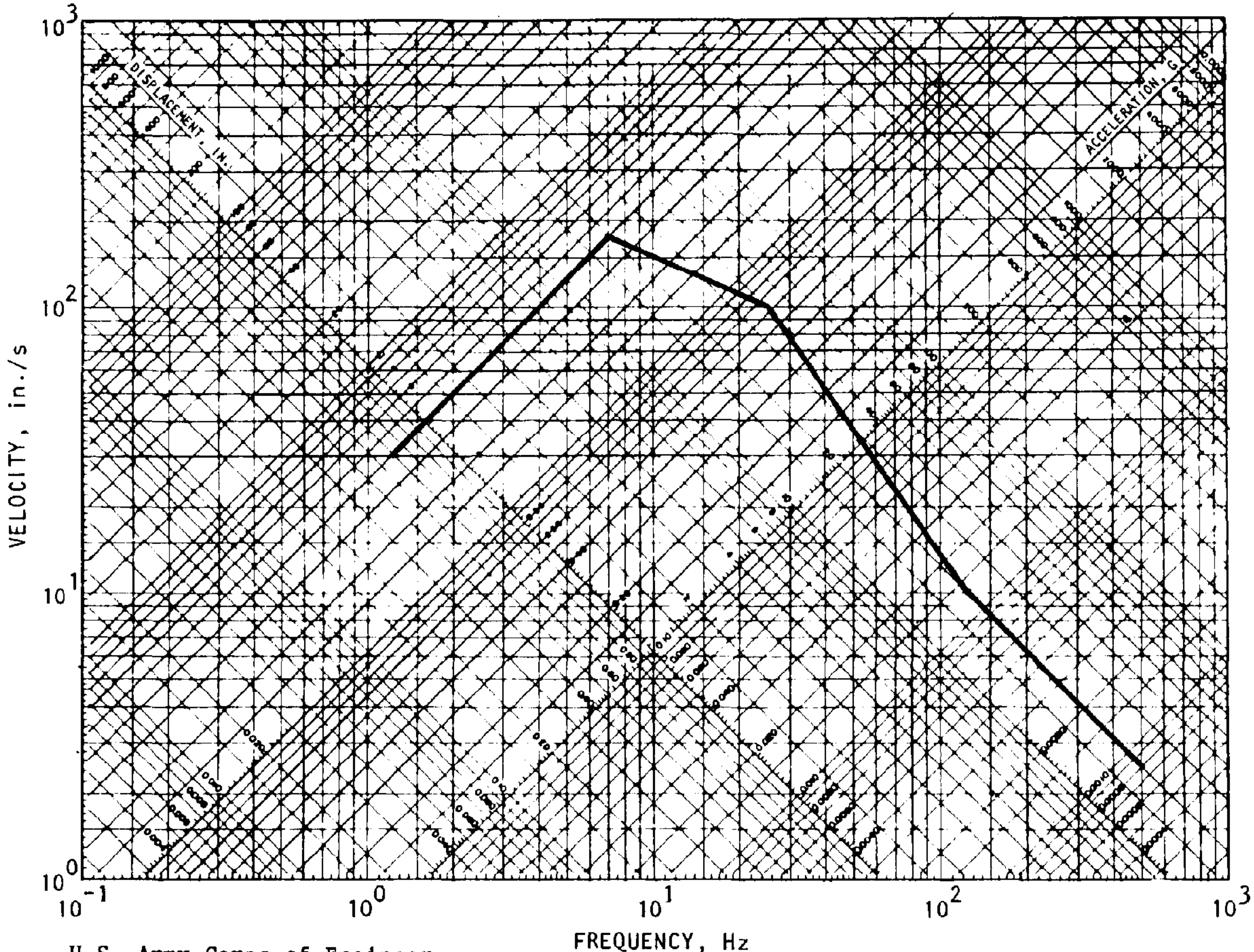

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FIGURE A-3CC-2. VERTICAL SHOCK SPECTRUM FOR OUTSIDE AIR MAKE-UP, LOCAL INSTRUMENTS, PRESSURE SWITCHES, PNEUMATIC CONTROL VALVE, PNEUMATIC ACTUATOR, AND TEMPERATURE TRANSMITTER

A-4. Shock Spectrum Analysis for Command Communication Equipment.

A-4-1. Nature of the survey. An intensive effort was undertaken to identify, describe, inspect, and assess a large group of command communication equipment items. Manufacturers were visited, commercial literature was received, and equipment was observed. Supplemental information was obtained from the Minuteman and Safeguard programs. Estimates of equipment fragility levels were established in the form of shock spectra.

A-4-2. Equipment description. Table A-4-1 provides a complete listing of the 75 pieces of equipment that were reviewed. The table includes manufacturers and model numbers as well as dimensions and equipment descriptions.

A-4-3. Shock spectrum criteria and assumptions.

a. The shock spectrum analysis provides a means of comparing many different time histories for damage potential based only on the assumption that the higher the shock spectrum, the greater the damage potential. The term Q is used to define the damping associated with the spectrum analysis. Q is related to the oscillator percent of critical damping as follows: $Q = \frac{1}{2}\zeta$. So for $\zeta = 10$ percent (or 0.1), $Q = 5$. The $q = 5$ damping is used because this represents a realistic value of damping for this type of equipment.

b. Definitions of the $Q = 5$ shock spectra fragility threshold levels derived for the command communications equipment items are provided in figure A-4-1. In establishing these threshold shock spectra levels, several assumptions were made:

(1) The levels were established for the floor/equipment interface. For equipment normally rack-installed as a drawer (or on top of a console), structural stiffness and packaging of the rack/console were assumed to be similar to other interfacing equipment.

(2) Floor tie-downs are required for all equipment installations, whether they are handmounted, or shockmounted, floor mounted or rack/console mounted.

(3) Cable interfacing requirements must also be addressed. These include source power and interface cable routing, relative displacements, tie-down, and securing techniques.

(4) In the case of the battery plant (telephone central office) the threshold level is estimated for the batteries in a tied down configuration. It was judged that the battery stacking support structure must be designed to withstand whatever personnel capability level is selected for further assessments, based on whatever battery weight/stacking configuration is considered.

A-4-4. Shock tolerance data.

a. Insufficient data exists for accurate shock and vibration vulnerability assessments of any of the command communication equipment identified as part of this study. That is, the actual shock and vibration vulnerability thresholds of these equipment items are unknown and will remain so until adequate tests are performed.

b. Because of the lack of sufficient shock and vibration vulnerability data, assessments of the C-E equipment were based on the limited amount of test data for similar equipment (e.g.: Minuteman, Safeguard, etc.) combined with engineering judgment and visual assessments of the equipment when possible. For those equipment items that were designed for military usage, the specific MIL-STD test specification referenced for that item was assumed to be representative of the highest level to which that item was capable of surviving. This may seem to indicate conservative estimates (low) of the equipment susceptibility; however, the lack of substantiating data from test reports dictated this approach. Visual assessments of some of this military equipment and supporting data from Minuteman also supported this conclusion.

c. Shock tolerance level assessments for each of the command communications equipment items are summarized in the right-hand column of table A-4-1. These levels are defined in figure A-4-1.

A-4-5. References and credits.

Boeing Aerospace Co. (BAC). *Personnel and C-E Equipment Shock Tolerance: Summary Report*, 2-1194-9100-164. Seattle, WA: BAC, Sep 1977.

TABLE A-4-1. COMMAND COMMUNICATION EQUIPMENT DESCRIPTION

EQUIPMENT ASSEMBLIES/ SUB-ASSEMBLIES	MODEL, TYPE, OR AN/ NO.	MANUFACTURER	DIMENSIONS				PART DESCRIPTION/USAGE	ESTIMATED FRAGILITY THRESHOLD LEVEL
			HEIGHT ~ CM (IN)	WIDTH ~ CM (IN)	DEPTH ~ CM (IN)	WEIGHT ~ KG (LB)		
TELEPHONE EQUIPMENT								
CROSS BAR SWITCH	758	WESTERN ELEC					4 WIRE AUTOVON DEDICATED SWITCHING	L3H, L4V
CROSS BAR SWITCH	304/306	WESTERN ELEC.					4 WIRE AUTOVON DEDICATED SWITCHING	L3H, L4V
TELEPHONE CENTRAL OFFICE	AN/FTC-31	PHILCO					ELECTRONIC TELEPHONE SWITCHING;	L1
POWER SUPPLY GROUP	OP-5/FTC		175(69)	53(21)	61(24)	272(600)	PERMANENT INSTALLATION.	L1
CONTROL GROUP	OK-9-FTC		175(69)	53(21)	61(24)	136(300)	ONE UNIT EACH OF EACH GROUP FOR EVERY 100 LINES	L1
SWITCHING MATRIX GROUP	OA-8181/FTC		175(69)	53(21)	61(24)	250(550)	(EXCEPT SWITCHING MATRIX GROUP WHICH REQUIRES	L1
REPEATER-REGENERATOR GROUP	OA-8182/FTC		175(69)	53(21)	61(24)	136(300)	ONE UNIT FOR EACH 50 LINES)	L3
MAIN DISTRIBUTION FRAME			175(69)	53(21)	61(24)	91(200)		
ALARM-FAULT INDICATOR	02-135/FTC		15(6)	25(10)	10(4)	13(30)		
TELEPHONE CENTRAL OFFICE	AN/TTC-38	GTE SYLVANIA	210(83)	220 (87)	373 (147)	2640 (5818)	ELECTRONIC TELEPHONE SWITCHING: RELOCATABLE	L4V, L3H
POWER SUBSYSTEM	1						INSTALLATION IN S-280 SHELTER, TRANSPORTABLE BY	L4V, L3H
OPERATOR SUBSYSTEM							TRUCK OR HELICOPTER.	L4V, L3H
COMMON CONTROL SUBSYSTEM							300 OR 600 LINE CONFIGURATIONS AVAILABLE.	L4V, L3H
NETWORK TERMINAL SUBSYSTEM								
ELECTRONIC SWITCH		WESTERN ELEC.					ELECTRONIC TELEPHONE SWITCH, ESS-TYPE	L2
TELEPHONE SETS	TA-341/TT	WESTERN ELEC.					2 WIRE/4 WIRE 12 KEY DESK SETS	L3
SWITCHBOARD CONSOLE	AN/GCC-11	PHILCO	155(61)	112(44)	102(40)		SWITCHBOARD USED WITH AN/FTC-31	L2
BATTERY PLANT	105D, E	WESTERN ELEC.	107(42)	58(23)	36(14)		BATTERY WITH RECTIFIER - POWER SUPPLY FOR TELEPHONE SWITCHING	L5
MULTIPLEX EQUIPMENT								
MULTIPLEXER	TD-1192 AN/FRC-98	TRW, INC.					ANALOG TO DIGITAL CONVERTER --MULTIPLEX AND DEMULTIPLEX 4 kHz ANALOG VOICE FREQUENCY	L3
MULTIPLEXER	TD-1193	TRW, INC.					STILL IN DEVELOPMENTAL STAGES	L3
DIGITAL CONVERTER	CV-3034	MAGNAVOX	9(3.5)	44(17.5)	41(16)	8.6(19)	VOICE DIGITIZER, REGENERATIVE DIGITAL REPEATER	L3

1 EACH OF THE 4 SUBSYSTEMS CONTAINS MANY COMPONENTS, EACH WITH AN INDIVIDUAL PART NUMBER.

TABLE A-4-1. (CONTINUED)

EQUIPMENT ASSEMBLIES/ SUB-ASSEMBLIES	MODEL, TYPE, OR AN/ NO.	MANUFACTURER	DIMENSIONS HEIGHT~ CM (IN)	WIDTH ~ CM (IN)	DEPTH ~ CM (IN)	WEIGHT~ KG (LB)	PART DESCRIPTION / USAGE	ESTIMATED FRAGILITY THRESHOLD LEVEL
TELEVISION EQUIPMENT								
CLOSED CIRCUIT TV CAMERA	650	DAGE CORP.	15(6)	10(4)	25(10)	2.3(5)	RECORDING OR TRANSMITTING VIDEO PRESENTATIONS	L3
CLOSED CIRCUIT TV MONITOR	RNCA9	CONRAC	20(8)	20(8)	36(14)	3.6(8)	PRE-RECORDED OR IN-PLANT VIDEO PRESENTATIONS OR SURVEILLANCE	L2
VIDEO TAPE RECORDER	VTP370	CONCORD	18(7)	36(14)	33(13)	18(40)	RECORDING UNIT FOR VIDEO PLAYBACK TAPES	L3
AUTOMATIC DATA PROCESSING EQUIPMENT								
CENTRAL PROCESSOR	6060	HONEYWELL	207(81)	171(67)	74(29)	566 (1250)	PROCESSES SOFTWARE INSTRUCTIONS - COMMUNICATES ONLY WITH SYSTEM CONTROLLER	L2
REMOTE TERMINAL INTERFACE	DATA NET 355		207(81)	285(113)	74(29)	954 (2100)	INTERFACE BETWEEN REMOTE TERMINALS AND CENTRAL PROCESSOR	L2
SYSTEM CONTROLLER	6060		194(76)	85(34)	74(29)	366 (850)	PROVIDES CONTROL OF AND COMMUNICATION BETWEEN ACTIVE SYSTEM MODULES	L2
LINE ADAPTER	355		194(76)	127(50)	74(29)	366 (850)	FOR DATA NET 355	L2
CARD READER AND CONTROL	CR2201		137(54)	102(40)	79(31)	386 (850)	READS PUNCHED CARDS FOR INPUT THROUGH THE I/O MULTIPLEXER	L3
CARD PUNCH AND CONTROL	CPZ 201		137(54)	112(44)	80(31.5)	386 (850)	HIGH SPEED CARD OUTPUT DEVICE DIRECTLY CONNECTED TO I/O MULTIPLEXER	L3
PRINTER	PRT 301		135(53)	160(63)	114(45)	683 (1500)	HIGH SPEED PAPER (PRINTED) OUTPUT DEVICE CONNECTED TO I/O MULTIPLEXER	L2
PRINTER CONTROLLER	PRT301		155(61)	115(45)	78(31)	453 (1000)	SEPARATE BUFFERED CONTROL FOR PRT 301 PRINTER	L3
MAGNETIC TAPE DRIVE	MTH501		166(65)	76(30)	76(30)	340 (750)	MAGNETIC TAPE READ (WRITE UNIT)	L2
TAPE DRIVE CONTROLLER	MTH501		128(50)	137(55)	70(27.5	518	CONTROLS ACTIVITY OF MAGNETIC TAPE READ/WRITE UNIT	L2
DISK DRIVE	DSS190		97(38)	56(22)	129(51)	317 (700)	DISK PACK READ/WRITE UNIT - 19 READ/WRITE SURFACES AVAILABLE	L3
DISK DRIVE CONTROLLER	DSS190		128(50)	137(55)	70(27.5	450 (650)	CONTROLS ACTIVITY OF DISK PACK READ/WRITE UNIT	L2
CORE MEMORY	TYPE 655		194(76)		74(29)		CENTRAL PROCESSOR MEMORY	L2
INPUT/OUTPUT MULTIPLEXER	6060	HONEYWELL	2.07(81)	1.16(46	75(29)	522 (1150)	STORED PROGRAM DEVICE CONTROLLED BY AND SHARING MEMORY WITH PROCESSOR	L2

TABLE A-4-1. (CONTINUED)

EQUIPMENT ASSEMBLIES/ SUB-ASSEMBLIES	MODEL, TYPE, OR AN/ NO.	MANUFACTURER	DIMENSIONS HEIGHT~ CM (IN)	WIDTH ~ CM (IN)	DEPTH ~ CM (IN)	WEIGHT~ KG (LB)	PART DESCRIPTION / USAGE	ESTIMATED FRAGILITY THRESHOLD LEVEL
CRYPTO EQUIPMENT								
ENCRYPT-DECRYPT UNIT	TSEC/KG-13	[1]	[1]				KEY GENERATOR INTERMEDIATE SPEED 24 BAUD RECEIVE AND TRANSMIT	L8
	TSEC/KG-34							L8
	TSEC/KG-81						CRYPTO FOR USE WITH TD-1192 MULTIPLEXER	L4
	TSEC/KW-7						CRYPTO UNIT - LOW SPEED 75 BAUD RECEIVE AND TRANSMIT	L8
	TSEC/KW-26						CRYPTO UNIT - LOW SPEED - 75 BAUD RECEIVE OR TRANSMIT	L4
	TSEC/KY-3						WIDE BAND ENCRYPTION - 50 KBITS RECEIVE AND TRANSMIT	L8
	TSEC/HY-11						A TO D CONVERTER - 9.6 KBITS - RECEIVE AND TRANSMIT	
TECHNICAL CONTROL EQUIPMENT								
SF UNITS	TM11-5805-667	DATA PRODUCTS INC.	12 (4.75)	32 (1.25)	37 (14.5)	2.3 (5)		L4
CIRCUIT CONDITIONING UNITS		DATA PRODUCTS INC.	49 (19.25)	44 (17.5)	13 (5)	4.6 (10)		L2
LINE AMPLIFIERS	DLA-3	DATA PRODUCTS INC.	11 (4.5)	32 (1.25)	37 (14.5)	2.3 (5)	COMPENSATES FOR LINE LOSS INCURRED DURING TRANSMISSION	L2
REMOTE ALARM UNITS	ALM-3	STELMA, INC.	9 (3.5)	48 (19)	18 (7)	2.3 (5)	SYSTEM ALARM AND FAULT INDICATOR	L2
REMOTE ALARM	ALM-3	STELMA, INC.					ALARM AND SYSTEM FAULT INDICATOR	L2

[1] INFORMATION ON AUTHORIZED MANUFACTURERS AND DIMENSIONS OF CRYPTO EQUIPMENT IS CLASSIFIED

TABLE A-4-1. (CONTINUED)

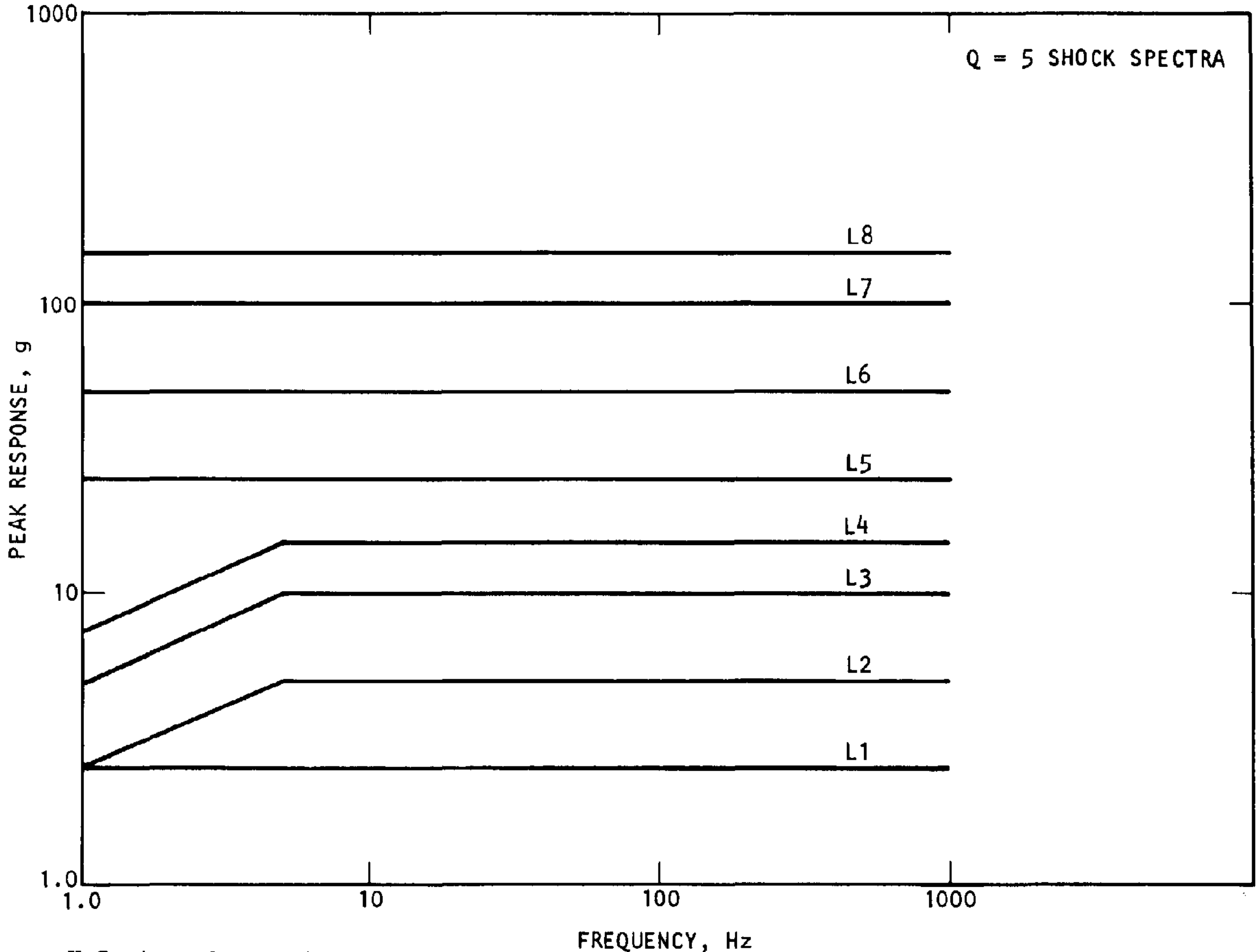
EQUIPMENT ASSEMBLIES/ SUB-ASSEMBLIES	MODEL, TYPE, OR AN/ NO.	MANUFACTURER	DIMENSIONS HEIGHT-CM (IN)	WIDTH-CM (IN)	DEPTH-CM (IN)	WEIGHT-KG (LB)	PART DESCRIPTION / USAGE	ESTIMATED FRAGILITY THRESHOLD LEVEL
MESSAGE PROCESSING EQUIPMENT								
CENTRAL PROCESSING UNIT	U90/60	UNIVAC	163(64)	163(64)	66(26)	527 (1160)	PROCESSES SOFTWARE INSTRUCTIONS; DIRECTS ACTIVITY OF PERIPHERALS (DISC, MAG. TAPE)	L2
TAPE STORAGE DRIVE	UNISERVO 16, TYPE 0862		163(64)	75(29.5)	72(28.5)	440 (975)	MAGNETIC TAPE/READ/WRITE UNIT	L2
TAPE FILE CONTROLLER	TYPE 5017		163(64)	95(37)	66(26)	318 (700)	CONTROLS ACTIVITY OF MAG. TAPE READ/WRITE UNIT	L1
DISC STORAGE DRIVE	TYPE 8425		102(40)	76(30)	61(24)	188 (415)	DISC PACK READ/WRITE UNIT (HANDLES 10-LEVEL DISCS)	L3
DISC FILE CONTROLLER	TYPE 5024		163(64)	95(37)	66(26)	318 (700)	PROVIDES AC POWER TO DISC DRIVES - CAN SERVE UP TO 8 DISC DRIVES	L2
CARD READER	TYPE 0716		109(43)	137(54)	83(32.5)	386 (850)	CARD READER-UP TO 1000 CARDS/MINUTE	L3
PRINTER	TYPE 0768		148(58)	132(52)	87(34)	635 (1400)	HIGH SPEED PRINTER	L2
INPUT/OUTPUT CABINET	TYPE 3024		163(64)	122(48)	66(26)	349 (770)	INTERFACE BETWEEN I/O PERIPHERALS (DISCS, MAG. TAPE, ETC.) AND PROCESSOR	L2
SYSTEM CONSOLE	TYPE 4014		136(53)	112(44)	71(28)	284 (625)	OPERATOR CONSOLE TO DIRECT I/O OPERATIONS AND DISPLAY SYSTEM PROBLEMS	L3
CONSOLE PRINTER	TYPE 0772		116(42)	86(34)	83(325)	181 (400)	PROVIDE PRINTOUT OF SYSTEM OPERATIONS, JOB NUMBERS AND JOB STATUS STATEMENTS	L2
POWER CONTROL CABINET	TYPE 3024		163(64)	173(68)	66(26)	619 (1365)	DIAGNOSTIC CENTER FOR INTERNAL SYSTEM MALFUNCTIONS	L2
STANDARD REMOTE TERMINAL	UNISCOPE 200		33(13)	46(18)	70(27)	41(91)	REMOTE TERMINAL FOR ACCESSING CENTRAL PROCESSING UNIT - JOB SUBMITTAL	L3
TELETYPEWRITER	MODEL 40	TELETYPE CORP.	48(19)	43(17)	64(25)	32(70)	SEND/RECEIVE KEYBOARD AND DISPLAY	L2

TABLE A-4-1. (CONCLUDED)

EQUIPMENT ASSEMBLIES/ SUB-ASSEMBLIES	MODEL, TYPE, OR AN/ NO.	MANUFACTURER	DIMENSIONS HEIGHT CM (IN)	WIDTH CM (IN)	DEPTH CM (IN)	WEIGHT KG (LB)	PART DESCRIPTION/USAGE	ESTIMATED FRAGILITY THRESHOLD LEVEL
RADIO EQUIPMENT								
RADIO RECEIVER	AN/GRR-24	ITT	8.9 (3.5)	48.5 (19)	34.5 (12)	10 (22)	SINGLE CHANNEL GROUND-TO-AIR UHF RECEIVER	L3
RADIO RECEIVER	AN/FRR-77	WESTINGHOUSE						
RECEIVER	R-1395/FR		18 (7)	48 (19)	55(21.5	32 (70)	RECEIVER TUNED TO ACCEPT TRANSMISSION 14-60 KHz	L4
DEMODULATOR-DATA CONVERTER	MD-1677/FR		13 (5)	48 (19)	55(21.5		DATA CONVERSION FOR PRINTOUT BY TELEPRINTER	L4
CODE SELECTOR	C-6941/FR		13 (5)	48 (19)	27(10.5	14 (30)	INTERFACE B/W RECEIVE SYSTEM AND CRYPTO	L4
TELEPRINTER	TT-513/FR			24(9.5)			PRINTOUT FROM DIGITAL INTERFACE	L4
TOTALS [1]			198(78)	234(92)	66(26)	903 (1990)	TOTALS FOR A COMPLETE AN/FRR-77 INSTALLATION	
LARGE SCREEN PROJECTION EQUIPMENT								
LIGHT VALVE PROJECTOR	PJ6000	GENERAL ELECTRIC	56(22)	43(17)	77(30.5	59 (130)	LARGE SCREEN TELEVISION PROJECTION, SINGLE GUN, SEALED LIGHT VALVE	L2
IMPROVED EMERGENCY MESSAGE AUTOMATIC TRANSMISSION SYSTEM (IEMATS) EQUIPMENT								
TERMINAL PROCESSING UNIT	[2]	BURROUGHS	213 (84)	102 (40)	91 (36)	521 (1150)	MESSAGE PROCESSING - CONTAINS INTERPRETER, DISC DRIVE, 64K MEMORY, POWER SUPPLY	L3
MASTER DISPLAY CONSOLE			71 (28)	66 (16)	104 (41)	27 (60)	CONSOLE FOR DISPLAY OF INCOMING/OUT - GOING MESSAGES - DESK TOP	L3
PRINTER			46 (18)	71 (28)	71 (28)	45 (100)	300 LPM PAPER PRINTOUT - DESK TOP INSTALLATION	L3
OPERATOR'S CONSOLE			76 (30)	122 (48)	76 (30)	91 (200)	FLOOR STANDING CONSOLE - TO DIRECT PERIPHERAL OPERATIONS	L3
SLAVE DISPLAY UNIT			51 (20)	66 (26)	61 (24)	18 (40)	CRT DISPLAYING DUPLICATE OF MESSAGES APPEARING ON MASTER DISPLAY	L3
PAPER TAPE READER/PUNCH			30 (12)	53 (21)	35 (14)	23 (50)	PAPER TAPE READ/WRITE UNIT - DESK TOP OR DRAWER INSTALLATION	L3
MAGNETIC TAPE UNIT			25 (10)	53 (21)	51 (20)	23 (50)	MAGNETIC TAPE READ/WRITE UNIT	L3
CARD READER			71(28)	71(28)	66(26)	48(105)	300 CARD/MINUTE READER - DESK TOP	L3

[1] TOTALS FOR AN ENTIRE AN/FRR-77 INSTALLATION.

[2] THE BURROUGHS CORP. WILL BE RELEASING THESE MODEL NUMBERS WITHIN THE NEXT FOUR WEEKS.



U.S. Army Corps of Engineers

FIGURE A-4-1. COMMAND COMMUNICATIONS EQUIPMENT TOLERANCE LEVELS

A-5. Shock Testing for Emergency Operating Centers in Albany and Oklahoma City.

A-5-1. Albany.

a. Suppliers of equipment to the emergency operating center (EOC), which is designed to function as the alternate seat of state government in Albany, were required to certify their installations for shock resistance or provide suitable isolation (Rempel, 1967). The standard of performance furnished by the owner was a trapezoidal response spectrum. Peak vertical speeds and displacements in this spectrum were just under 20 in./s and 5 in., respectively. Maximum vertical and horizontal acceleration were taken as 18.8 *g*. Although peak horizontal speeds and displacements were somewhat less than the values for the corresponding horizontal parameters, equipment was required to meet the same standard in all axes. The two hammer-blow machines (LWHI for lightweight and MWHI for medium weight equipment) were used for the bulk of the testing. The Air Force drop test (VDMI) supplied some of the data.

b. For items not suitable for attachment to the standard testing machines, shock tests were improvised. A heavy, bulky motor generator was put on a railroad flatcar and rolled into stationary cars to provide a simulation of horizontal shock motion; the two ends were then dropped individually onto wooden blocks to give vertical shocks.

c. In some cases shock testing was done to help the design of isolation, that is, to provide a known safe level of impact below the expected maximum. Some suppliers could not certify their equipment above the level needed for ordinary long distance shipment (which is usually taken to mean a peak acceleration of 3 g) or some other previously established peak acceleration.

d. A summary of the test results is provided in table A-5-1. Table A-5-2 gives a brief list of equipment whose manufacturers certified tolerance levels below the expected maximum.

A-5-2. Oklahoma City.

Somewhat similar requirements appear to have been laid down for the city and state EOC's in Oklahoma City. Some of the results appear in table A-5-3.

A-5-3. Conclusions.

a. Two noteworthy points emerging from the test documentation connected with the three EOC's are: the ruggedness and suitability for hard mounting of much equipment ranging in size from lighting fixtures to 100 HP motor-pump sets; and the possibility of modifying equipment as deficiencies are found. Although the diesel generator was fairly seriously impaired during testing, the source of the failure under such loading was easily located and removed. The motor controller cabinet showed a resonance at 9 Hz, but the structure was simply modified so that the resonant frequency became 25 Hz (which frequency was more amenable to testing). Various mounts that broke under test were strengthened.

b. In contrast to the military tests of a fan reported above, all the fans in tables A-5-1 and A-5-2 passed the tests applied, one of which was a severe hammer blow.

c. As noted in the tables, there are two or three items for which the lack of strong low-frequency components in the test motion to simulate the strongest likely ground-transmitted motion might easily be important: the two waterchilling systems, a 10-ft-high electric distribution panel, electric switch gear assembly, and a 9-ft-high motor control center. There may be other items sensitive in this way, such as the 7 by 7-ft air filter and possibly the large motor-pump sets, the two components of which are connected by a long, relatively thin shaft.

A-5-4. References and credits.

Rempel, J-R. Ground Shock and the Contents of Personnel Shelters *Resistance of Human and Inanimate Contents of Hardened Shelters to Nuclear Induced Motion*. Menlo Park, CA: Stanford Research Institute, Nov 1967.

TABLE A-5-1. SUMMARY OF SHOCK TESTING FOR BOC, ALBANY, N.Y.

Description of Equipment	Size	Manufacturer/Manufacturer's Designation	Description of Test	Result	Comment
1. Annunciator	3' x 4' x 1/2' (?)	Panalarm Division of Panellet, Inc., Skokie, Ill. Model 5127 (125 v DC)	LWHI, 11" drop, side to side, 25 g max 5-1/2" drop, front to back, 19 g max 19" drop, top to bottom, 80 g max*	No damage or operational impairment	Test adequate for airslap and probably also for all components of ground motion since equipment is small
2. Annunciator	As above	Panalarm Division of Panellet, Inc., Skokie, Ill. Model 5136 (115 v AC)	LWHI, 11-1/2" drop, side to side, 19 g max, 9" drop, front to back, 28 g max 22 " drop, top to bottom, 19 g max	Panel light operation unimpaired Mechanical damage: 3 indicators fell off	As above Mechanical damage not serious
3. Auto Roll Unit with Electric Motor (Air Conditioning Equipment)	110 v 1 ϕ ~ 5' x 6' x 2'	Electro-Air Cleaner, Inc.	MWHI, 1' drop, 2' drop } 100 g max, vert. only (?)	Base of filter drive motor broke: otherwise no damage	Because of size, equipment may have low modal frequencies; test probably inadequate for strong ground-transmitted motion
4. Batteries (2) Ni-Cd	15" x 15" x 6"	Nife, Inc., Copiague, N.Y. KB1-25	LWHI, 12" drop, Y-axis, 3" drop, X-axis, 3-1/2" drop, Z-axis } 20+ g max	No damage	Why do 3" drops produce same acceleration of same equipment as 12" drops? Doubt peak g measurement applied to whole equipment.
5. Centrifugal Fan	2700 lb	Bayley Blower Co., Milwaukee, Wis. Model SC-TH	Rolling table for horizontal test; speed not reported Drop for vertical, height not reported 7 g max (?), both axes	No damage or operational impairment	From data given, adequacy at test not conclusive.
	1575 lb	Bayley Blower Co., Milwaukee, Wis. Model BC-UB	Direct drive mechanical vibration machine 7 g max, 3 axes	Same as above	Same as above
	1175 lb	Bayley Blower Co., Milwaukee, Wis. Model BC-TH	Same as above	Same as above	Same as above
	850 lb	Bayley Blower Co., Milwaukee, Wis. Model BC-UB	Same as above	Same as above	Same as above
	250 lb	Bayley Blower Co., Milwaukee, Wis. Model BC-UB	Same as above	Same as above	Same as above
6. Centrifugal Pump and Motor	total weight 1650 lb overall length ~5' 60 hp, 1750 rpm 3 ϕ, 60 Hz	Allis Chalmers, Milwaukee, Wis. Model 211-494-502 Size 5 x 4 Tupe SJ Allis Chalmers, Norwood, O. NEMA Des. B Type AP	MWHI, 1.25' and 2.25' drops, vertical only (?)	No operational impairment Weld cracked open on mounting foot	Vertical modes probably adequately tested for airslap
7. Compressor and Motor	3-1/2" x 2-1/2" 3/4 hp 1750 rpm 115/230 v 1 ϕ, 60 Hz	Quincy Compressor Co., Quincy, Ill. A4 377955 Century Electric Co., St. Louis, Mo. CS 68 FHK3-3FA	LWHI, 20 g peak, all axes	No damage	Adequate for airslap, probably also adequate for all likely motion of this small equipment
8. Converter (Heater)	150 psi, 375° F 6' x 1' dia.	Taco Heaters Inc., Cranston, R.I. 142125R-B	MWHI, 19 to 23 g peak* through 80 Hz low pass filter, 3 axes	Mounting bolts severely bent during blows parallel longitudinal axis No operational impairment	Probably adequate test for airslap. Behavior at low frequencies not established.
9. Distribution Panel (with Circuit Breakers)	10' x 4' x 1' (?)	General Electric Co., Schenectady, N.Y. Panel "A"	MWHI, 3" drop, vertical, 20 g max† 3" drop, side, 20 g max† 6" drop, back, 22.5 g max†	No damage or operational impairment	This large equipment should be tested for resistance to ground-transmitted motion also (Equipment had slight modifications for shock service.)
10. Distribution Panel (with Circuit Breakers)	5' x 4' x 1' (?)	General Electric Co., Schenectady, N.Y. Panel LP-ULB	MWHI, 2-3/4" drop, vertical 21.5 g max† 4" drop, side, 20 g max† 6" drop, back, 22 g max†	Same as above	Same as above
11. Light Fixtures: Lightalarm Battery Charging Unit Recessed Downlight Vaportite Ceiling Unit Vaportite Bracket Turnlox Reflector Unit Compact Directionals Compact Directionals 40 W Rapid Start Weatherproof Bulletin Luminaire Skylouver		General Electric Co., Schenectady, N.Y. WP402NCVA 4-601A McPhilben N-43-45 VT McPhilben N-43-40 VT McPhilben G-7642 Benjamin SOW-6M McPhilben SOE-6M McPhilben A1 Benjamin K-44981 Simes CFR Guardian 2047 LPF-TS Electrolight	MWHI, vertical or Z—4" drop, 22 g max on fixture LWHI, horizontal or X and Y—various drops 4" to 13", 30+ g max	No damage except to Bulletin Luminaire, which must be shock mounted Heavy duty bulb required in 40 W Rapid Start	Probably adequate test for this equipment except for possible swing modes in suspended ceiling fixtures
12. Motor Control Center	44" x 90-3/8" x 20-1/4" 1900 lb 440 v operating 110 v control	Westinghouse Electric Co., St. Louis, Mo. Class 11-350	VDMI, Free fall into sand with maple blocks as stoppers, 7-1/2" fall, 50 g max, vertical 20 g max, horizontal Frequency search	Angle clip holding control device panel broke; corrected by redesign Tendencies for two breakers to trip on vert. blow; corrected by choosing larger breaker Tendency for momentary closure of breakers on transverse blows; corrected with anti-shock latch Otherwise no damage Equipment has frequency at 9 Hz; changed to 25 Hz by redesign	Device as modified adequately tested for airslap

LWHI = lightweight high impact test machine MWHI = medium weight high impact test machine VDMI = variable duration, medium impact test machine

TABLE A-5-1. (CONCLUDED)

Description of Equipment	Size	Manufacturer Manufacturer's Designation	Description of Test	Result	Comment
13. Motors (Electric) 1800 rpm	 10 hp 15 hp 25 hp 75 hp	Not reported SK4256A21 SK4284A22 SK4324A21 SK4405A2	All dropped 6" in normal position onto steel plate: 189 g max through 25 Hz low pass filter 1100 g max through 2000 Hz low pass filter For horizontal tolerance motors mounted on channels, which were wheeled into shock mounts on wall: 7 g max through 25 Hz low pass filter both horizontal directions, 20 in./sec speed change	No damage nor operational impairment	Horizontal lists barely adequate for airslap but equipment probably withstands all likely ground motion
14. Motor-Generator (Diesel) 4 cycle 6 cylinder 9" x 10-1/2" bore and stroke 900 rpm, 60	40,000 lb 580 Bhp 500 KVA 400 KW	Chicago Pneumatic Tool Co., N.Y., N.Y. 69-CPHS	RR car impact, 12 MPH, 7 g max, longitudinal and transverse 23" free fall at motor end only, 6.6 g max 19" free fall at generator end only 7 g max Duration of deceleration 1 to 2 sec	Overheating due to fouled lube line (impact loosen scale and weld splatter); corrected by stainless steel line and cleaning Gage gloss loosened-not disabling No observable permanent bending or lessening of clearances	Good test for airslap at ~25 psi, probably not severe enough for 50 psi or for strong ground-transmitted motion in all environments.
15. Motor Sentinel	4-3/4" x 2-3/4" x 2-3/4" 1 lb	Westinghouse Electric Corp., Beaver, Pa.	VDMI, free fall into sand with maple blocks as stoppers. 13" free fall, 50 g max, all axes	No damage	Adequate for airslap, probably also adequate for all likely motion of this small equipment
16. Panelboards[§] MS-E2-L1 MS-E2-L2 Control Supply Tel. Cab. Tel. Cab. Tel. Cab.	 19-1/4" x 20" x 7-1/2" Same as above Same as above 36" x 18" x 4" 6" x 6" x 4" 28" x 10" x 4"	Westinghouse Electric Co., St. Louis, Mo.	VDMI, free fall into sand with maple blocks as arresters, 50 g peak, all axes	No damage; no false opening or closing	Tests probably adequate for both airslap and strong ground-transmitted motion due to small size of equipment. Should also be tested in racks.
17. Panelboards[§]	 19-1/4" x 20" x 7-1/2" 24-3/4" x 20" x 7-1/2" 24-3/4" x 20" x 7-1/2" 38-1/4" x 20" x 7-1/2" 44" x 20" x 7-1/2" Same as above	Westinghouse Electric Co., St. Louis, Mo. CCP PH-PE2 OCE-2 PH-PA CCA-Section #1 CCA-Section #2	VDMI, free fall into sand with maple blocks as arresters, 25 g peak, all axes	No damage; no false opening or closing	Same as above
18. Pump and Induction Motor	5 hp 1740 rpm 3ϕ, 60 Hz 150 gpm 42 ft TDH	Continental Electric Co., Rockford, Ill. WV 215C Food Mach. & Chem. Corp., Chicago, Ill. LMC-4	MWHI, drop height not reported 6 to 8 g max[†] through 80 Hz low pass filter, 3 axes	No damage or operational impairment	Reaction to frequencies below 20 Hz may not have been adequately tested (Shape of equipment: cylindrical ~5' x 8" dia.) Test not quite adequate for airslap above 50 Hz
19. Pump and Motor (Submersible)	2 hp 440 v 3ϕ, 60 Hz 20 gpm 4"	Franklin Electric Co., Inc. Model 3P1078B4D Peerless Pump Division Food Mach. & Chem. Corp. Indianapolis, Ind. Model 420D200	LWHI, drop not reported 20 to 23 g max[†] through 80 Hz low pass filter, 3 axes	No damage or operational impairment	Probably adequate for this equipment (Pump size ~36" x 4" dia.)
20. Pump and Motor	100 hp 3ϕ, 60 Hz 440 v 3450 rpm	Buffalo Pumps, N. Tonawanna, N.Y. Reliance Electric & Engineering Co., Cleveland, O.	MWHI, 2.25' drop } 100 g max, 1.25' drop } vertical only (?)	Performance unaffected Slight increase in mechanical unbalance in pump (?)	Because of rotational symmetry of equipment, vertical testing probably enough Adequacy of test below 20 to 30 Hz doubtful (Equipment probably has frequencies below 30 Hz)
21. Switchgear Assembly Hand-Operated Breakers Low-Voltage	 3 sections x 27" wide 4 sections x 20" wide 3 sections x 26" wide 4 sections x 20" wide	General Electric Co., Philadelphia, Pa. AKD-5 AK-2A-50 } breakers AK-2A-25 } AKD AK-2-50 } breakers AK-2-25 }	MWHI, 4" drop, 40 g max vertical 20 g max horizontal MWHI, 10" drop, 100 g max, vertical 50 g max, horizontal	No damage, closed breakers may open above 15 g	Probably adequate for airslap only. Cabinetry may have low frequency responses not explored
22. Transformer 3ϕ, 60 ~	225 KVA	General Electric Co., Schenectady, N.Y.	MWHI, 5" drops, 25 to 27 g peak	No damage	Probably adequate test for this equipment
23. Waterchilling System	12,300 lb (dry)	Borg-Warner, York, Pa. HT-24	Free fall onto springs,[Δ] 6-7/8" fall, 13/16" spring defl. 7 g max, vertical Pendulum swing into springs,[Δ] 63" to 70" swing, 9/16" to 17/16" spring defl., 7 g max both horizontal axes	No damage nor operational impairment	Not enough data to determine adequacy for this particular equipment, but test motion does not properly simulate either airslap or ground-transmitted motion; fragility level not established; this large equipment probably has frequencies especially susceptible to strong ground-transmitted motion
24. Waterchilling System	30,500 lb (dry)	Borg-Warner, York, Pa. MT 25	Free fall onto springs,[Δ] 6-5/8", 17/16" spring defl., 7 g max, vert. Pendulum swings into springs,[Δ] 66-1/2 to 70-3/8 swing, 9/16" spring defl., 7 g max both horizontal axes	Same as above	Same as above

[*] Measured at anvil; other readings on equipment.

[†] Location of accelerometer not reported.

[§] Panels were attached to shock platform through cupmounts made by Barry Controls, Inc., Cat. Nos. 1010, 1015, 1035.

[Δ] Vibration Mountings, Inc., Type SYA-20, 24,700 lb/in., 4 used with HT 24, 8 used with MT 25.

TABLE A-5-2. NOMINAL FRAGILITY LEVELS USED FOR EOC, ALBANY, N.Y.

DESCRIPTION OF EQUIPMENT	SIZE	MANUFACTURER/ MANUFACTURER'S DESIG.	STATED FRAGILITY LEVEL
1. Blower (garbage room)	92 lb	Brunner Div. Dunham Bush, Inc. WJ-45	10g, 3 axes (?)
2. Cabinet, hydraulic jack	23" x 16" x 7" 95 lb	Not reported	3g,
3. Cabinet, motor controller	108 lb ~ 28" x 13" x 14"	Mosler Safe Co., Hamilton, O. (?)	3g
4. Compressor (refrigerated garbage)	156 lb	Brunner Div. Dunham Bush, Inc. WC-50-FC	10g, 3 axes (?)
5. Dishwasher	1000 lb (inc. water)	Toledo TA-27	6g, 3 axes (?)
6. Hydraulic jack (blast door)	16 ton capacity 48" stroke 465 lb	Richard Dudgeon (?)	> 18.8g w/door closed and locked < 18.8g w/door open
7. Hydraulic pump	997 lb ~ 44" x 22" x 37"	Not reported	3g
8. Purge compressor	120 lb 11" x 11" x 18½"	York	≥ 1.7g vert.
9. Radiator	2100 lb 846 rpm	Young Radiator Co. model 1450	3.5g, 3 axes (?)
10. Range (electric)	390 lb	Hotpoint model HRG 13	3.5g, 3 axes (?)
11. Refrigerator and freezer	~ 1000 lb (loaded)	Victory Metal Manu. Co. model VS-48-S model VFS-18-S	3.5g, 3 axes (?)

TABLE A-5-3. SHOCK RESISTANCE CERTIFICATION FOR CITY AND STATE EOC'S, OKLAHOMA CITY, OKLAHOMA

DESCRIPTION OF EQUIPMENT	SIZE	MANUFACTURER/ MANUFACTURER'S DESIG.	DESCRIPTION OF TEST	RESULT	COMMENT
1. Air Filter	7' x 7'	American Air Filter Co. Louisville, Ky. 7-70 Roll-o-matic model B	MWHI, 1.25' drop, 3" table travel 2.25' drop, 1.5" table travel	No operational impairment Slight modification of structure required	Probably an adequate test even though low frequency response not explored
2. Fan w/electric motor (centrifugal)	12000 cfm 1650 lb	American Blower Corp. Detroit model no. C12A	MWHI, 1.5', 2.5', 3.5', 4.5', and 5' drops	Required stronger mount otherwise no operational impairment	Probably an adequate test even though low frequency response not explored
3. Pump and motor (centrifugal)	717 lb 10 hp	Westinghouse WS-107A-2 model no. 3655	VDMI 13" drop into sand, 3 axes 7 wooden blocks as arrestors	No observable damage (dye-penetrant test applied)	Peak accel. during test somewhat below maximum possible in field otherwise comment same as above

MWHI = medium weight high impact test machine

VDMI = variable duration, medium impact test machine

APPENDIX B

DESCRIPTION OF COMPUTER PROGRAMS

B-1. General. Computer programs are available for four areas relevant to the tasks of the shock isolation system designs:

—Wave form synthesis (par. B-2)

—Dynamic analysis (par. B-3)

—System optimization (par. B-4)

—Analysis of test data (par. B-5)

This appendix briefly describes the available computer programs in these areas. Tables provide quick-reference data such as program developer, language, system used, availability, and other specifications. It must be noted that certain details of less important programs were not readily available and are not included. However, the interested reader can contact the cited sources for missing details. For further information regarding availability and use of these programs, contact the U.S. Army Corps of Engineers, Huntsville Division, Huntsville, Alabama 35807.

B-2. Wave form synthesis.

a. Program purpose. The WAVESYN computer program has been formulated for synthesizing a time history to match a given shock response spectrum. The wave form component selected as the basic element of the synthesized time history is a sinusoid with an odd number of half cycles and an amplitude that varies continuously in time and is proportional throughout the duration of the time history to a half-cycle sinusoid. Table B-1 lists WAVSYN specifics.

b. Program features.

(1) Formulation techniques. The synthesis procedure may begin by filtering the time history from which the spectrum was generated, if such data are available, and defining the amplification ratios as functions of frequency. Input data to the computer program consists of spectrum frequencies and response amplitudes at which a match will be forced and the number of half cycles in each frequency component to achieve the measured or estimated amplification ratio. The computer program then calculates the element frequencies (not necessarily the same as the selected spectrum frequencies), adjusts the amplitude of each element so that the composite wave matches the spectrum at the selected points, and generates a response spectrum of the composite wave form at intermediate as well as at the selected frequencies.

(2) Usability. The wave form synthesis technique (WAVSYN) has been improved to include time phasing of the wave form components. Certain constraints on the phase relationship of the components are required to assure convergence of the solution and proper matching of the spectrum. Based on a semiempirical approach, rules for selecting optimum system frequency ratios were developed such that deviation between the spectrum generated by the synthesized wave form and the required spectrum would be minimized. An automatic method was added to the WAVSYN program to provide minor adjustment of the selected system frequency ratio and improve accuracy.

(3) Input parameters. Spectrum frequencies, response amplitudes, and control statements are used.

(4) Output parameters. Standard print/plot output is produced.

(5) References.

(a) Yang, R. C. *Development of a Wave Form Synthesis Technique*, SAF-64. Los Angeles: The Ralph M. Parsons Co., 28 Aug 1970.

(b) Yang, R. C. *Modification of the WAVESYN Computer Program*, SAF-82. Los Angeles: The Ralph M. Parsons Co., 30 Apr 1971.

(c) Yang, R. C. and Saffell, H. R. "Development of a Wave Form Synthesis Technique—A Supplement to Response Spectrum As a Definition of Shock Environment," *Shock and Vibration Bull.* 42, pt. 2, Jan 1972.

B-3. Computer programs for dynamic analysis (table B-1).

a. ISOL. ISOL (Dynamic Response for a Rigid Platform Six Degree of Freedom Shock Isolation System) calculates the dynamic response of rigid-mount shock isolation systems having six degrees of freedom, supporting rigid loads, and subjected to base excitations. The program comprises several important capabilities: elastic-pendulum, pivoted base-mounted and special types of shock isolation systems can be analyzed; linear or nonlinear spring and damper element characteristics can be specified; static equilibrium solution of the platform system is obtained. Multiple cases may be easily processed, and c.g.'s of the system can be readily selected. An option is available to plot the response time histories of all parameters.

(1) Formulation Technique. A finite element mathematical model is used to describe the platform and mounted equipment. The platform model uses elastic elements between node points. Linear or non-

TABLE B-1. BASIC SPECIFICATIONS FOR COMPUTER PROGRAMS USED IN ANALYSIS OF SHOCK ISOLATION SYSTEMS

Type	Code Name	Developer	Year	Deg of Freedom	Nonlinear Capabilities	Language	System, Series	Availability
Wave Form Synthesis	WAVSYN	R.C. Yang The Ralph M. Parsons Co. Los Angeles	1970-71	NA	NA	Fortran IV and control statements	CDC 6600 Univac 1100 Series	U.S. Army Eng. Div., Corps of Engineers Huntsville, AL 35805
Dynamic Analysis	ISOL	The Ralph M. Parsons Co.	1969	6/node	Spring and damper element characteristics	Fortran V	CDC 6600	U.S. Army Eng. Div., Huntsville
	ISIP	Sperry Rand Space Support Div., Huntsville, AL	1972	6/node	Spring and damper	Fortran IV	CDC 6400	U.S. Army Eng. Div., Huntsville
	ISOLIN	Sperry Rand Space Support Div.	1972	6/node	Damping and stiffness coefficients	Fortran IV	CDC 6400	U.S. Army Eng. Div., Huntsville
	SPECTRUM	Sperry Rand Space Support Div.	1972	NA	NA	Fortran IV	CDC 6400	U.S. Army Eng. Div., Huntsville
	FVDUS	The Ralph M. Parsons Co.	1969-70	6-rigid bodies 140-flexible bodies	Linear systems only	Fortran V	CDC 6600	U.S. Army Eng. Div., Huntsville
	ISIP2	Sperry Rand Space Support Div.	1971	140 for flexible platform systems	Flexibility and inertia matrices can be table read-in (springs must be linear)	Fortran IV	CDC 6400	U.S. Army Eng. Div., Huntsville
	TRI/SAC	Agbabian Associates El Segundo, CA (based on SAP Code developed under E.L. Wilson, UC Berkeley)	1976	6/node max DOF flexible	Limited to bar elements (other elements elastic), any combination	Fortran IV	CDC 6400 CDC 7600 Univac 1108	University Software Systems (subsidiary of Agbabian Associates) 250 N. Nash El Segundo, CA 90245
	STARDYNE	Mechanics Research Div., Systems Development Corp., Santa Monica, CA 2500	1972 (latest update)	Static analy.: Up to 2500 nodes at 6 DOF each. Other: Up to 999 nodes at 6 DOF	Linear elastic only	Fortran IV	CDC 6600	Control Data Corp.: All 6600 data centers
	NASTRAN	Computer Science Corp. MacNeal-Schwendler Corp., Martin-Marietta, Bell Aerosystems	Level 15.1-1972 15.5-1973 16 -1976	--	--	Fortran IV and machine-specific language	IBM 360/370 Univac 1108 CDC 6400/6500/6600	Computer Software Management and Information Center (COSMIC) 112 Barrow Hall University of Georgia, Athens, GA 30502
	MSC/NASTRAN	MacNeal-Schwendler Corp., Los Angeles	1972	--	--	Fortran IV	IBM 360/370 Amdahl series Fujitsu M series CDC 7600 CDC cyber series Univac 1100 series	MacNeal-Schwendler Corp. 7442 N. Figueroa St. Los Angeles, CA 90041 and Most major commercial data centers

linear elements are used to represent the isolators connecting selected node points to points of support on the building and building motions are input at these supports. A set of second-order differential equations is solved iteratively using numerical integration by simple finite differences to obtain the dynamic response.

(2) Usability. The mass points are limited to 50, the elastic/damper elements to 50, and the points of interest to 4. The program is highly versatile for rigid platform systems. Time is consumed calculating spring and damping element characteristics and the time motion history, which are assumed to be known. The output data must be converted to shock spectra form. ISOL is most effectively used for large motion problems.

(3) Input parameters. These are the control statements, the system coordinate data, pendulum and/or ball descriptions, spring and damper characteristics, and excitation tables.

(4) Output parameters. Components of relative displacement and absolute acceleration of the c.g. or other point of interest are tabulated for each time period.

(5) Reference. *Development of Standard Design Specifications and Techniques for Shock Isolation Systems*, Volumes I, II, and III, SAF-37. Huntsville, AL: U.S. Army Engineer Div., Corps of Engineers, 12 Feb 1970.

b. ISIP. The ISIP (Integrated Shock Isolation Program) program integrates several modular programs into one package to make the shock isolation analysis a continuous process. This integration eliminates problems related to data linkage between modular programs. A shock isolation system with a rigid platform can be analyzed for its dynamic response automatically.

(1) Formulation technique. A problem is executed by selecting the required options from the modular programs ISOLIN, NWVSYN, SYNSOL, ISOL, PLTSUB, SPCTRUM, FILTER, and FOURIE.

(2) Usability. The program looks promising but has yet to be confirmed for practicality. For the maximum number of spectrum frequency points, the limits are 14; for mass points, 50; for elastic damper elements, 50; for points of interest, 4; and for input excitation table, 500.

(3) Input parameters. These are a function of input requirements for modular programs; control indices allow usage of modular programs or an entire integrated shock isolation system analysis.

(4) Output parameters. These are also a function of modular programs, particularly PLTSUB (the plotting module). Time motion histories are available, as are plotted response spectra for mass points in the system.

c. ISOLIN. The ISOLIN program relieves the design engineer of the tedious work of preparing input data for the ISOL program. It includes computation of spring/damper constants for static equilibrium of platform and equivalent weight, c.g., and mass moment of inertia.

(1) Formulation techniques. These are the same as described in the preceding paragraph.

(2) Usability. This is a highly versatile program. The limits for mass points are 50; for damper elements, 50; and for points of interest on a platform, 4.

(3) Input parameters. These comprise the control statements, isolator system geometry, inertia data, and engineering design parameters of the platform.

(4) Output parameters. Print/plot output includes a plan view of platform; input data; load values, spring rate, unstretched length, and damping constant for each isolator; the isolator position to c.g.; the equivalent system weight; mass moments of inertia, system frequencies; and a list of punched output for the ISOL program.

d. SPCTRUM. The SPCTRUM (Shock Spectrum) program generates a shock response spectrum from a given acceleration-time motion history. It can use output from the ISOL program as input.

(1) Formulation technique. Input data are used to perform a numerical integration by the linear acceleration method to produce maximum displacements and accelerations at specified frequency points.

(2) Usability. This is program designed specifically for computing the shock spectrum at selected frequencies. It will plot complete shock spectra on a line printer.

(3) Input parameters. These comprise the control statements, frequency range and intervals, acceleration-time data, and shock analysis constants.

(4) Output parameters. Provided are the maximum displacements and accelerations as a function of time and specified frequency points. The response is plotted for system c.g. and four other specified points.

e. FVOUS. In the FVOUS (Evaluation of Response of Linear Shock Isolation Systems by Use of Response Spectra) program, the dynamic response of a linear shock isolation system is calculated by using a modal analysis method. The program will compute stiffness and mass matrices for rigid body systems; frequencies, mode shapes, and modal participation factors; and response motions for linear shock isolation systems.

(1) Formulation technique. A rigid or flexible platform shock isolation system is mathematically modeled by an assemblage of mass points and isolator elements. Then the spring constants, flexibility, and inertia matrices (for flexible systems), and frequency table must be described. FVOUS calculates a dynamic matrix for rigid body motions and response data for both rigid and flexible systems in terms of relative dis-

placements and absolute accelerations of each mass point. The dynamic responses are computed by a normal modal analysis, which sets up a "modal" matrix and general energy equations to compute the final responses.

(2) Usability. The program handles rigid or flexible platform systems. Its limits are, for mass points, 100; for springs, 100; for degrees of freedom for flexibility problems, 140; for normal modes, 20; and for points on shock spectrum, 20. FVOUS is effectively used for small motion problems.

(3) Input parameters. These comprise the control statements, mass data, spring constants, flexibility and inertia matrices for flexible bodies with more than six DOF; shock spectrum points; and frequency table.

(4) Output parameters. These include a printout of input: mass, stiffness, and dynamic matrices for rigid body motions; eigenvectors; participation factors; response data in terms of relative displacements; and absolute accelerations of the mass points.

(5) Reference. *Development of Standard Design Specifications and Techniques for Shock Isolation Systems*, Volumes I, II, III, SAF-37. Huntsville, AL: U.S. Army Engineer Div., Corps of Engineers, 8 Jan 1971.

f. ISIP2 PROGRAM. Using a normal modal analysis technique, ISIP2 (Flexible Platform Shock Isolation Analysis) calculates the transient response of a flexible platform shock isolation system to an input shock spectra.

(1) Formulation technique. A finite element model is defined using beams, nodes, and springs. A platform stiffness matrix is formulated for the model and then inverted to obtain a platform flexibility matrix. Nodal masses are used to define an inertia matrix and moments of inertia about X, Y, and Z axes. Resonant frequencies and mode shapes are then obtained by normal modal analysis. Finally, the input shock spectrum characteristics are used to compute the dynamic transient reponse in terms of relative displacements and absolute accelerations of each mass point.

(2) Usability. The program handles rigid- or flexible-body dynamics modes and 20 points on a shock spectrum. It is good for small motion problems.

(3) Output parameters. These include mass, stiffness, and dynamic matrices for rigid body motion; eigenvectors; participation factors; and relative displacement and absolute accelerations for each mass point.

(4) Reference. *Integrated Shock Isolation Design Manual*. Huntsville, AL: U.S. Army Engineer Div., Corps of Envineers, Mar 1972.

g. TRI/SAC. This system is a general-purpose digital computer program for three-dimensional analysis of structural systems using the finite element approach. The analysis types available include static response, dynamic response, response spectrum method, and a capability for the solution of certain nonlinear dynamic problems.

(1) Formulation techniques. TRI/SAC has three executable programs: the proprocessor (PRESAP), the main program (RSPNSE), and the postprocessor (SAPOUT). PRESAP provides a 3-D plot of input model, optimizes the bandwidth of stiffness matrix, provides a 3-D plot of optimized mesh, extracts mode shapes by Rayleigh-Ritz or inverse iteration methods, and plots 3-D mode shapes and data tape for input to RSPNSE. RSPNSE provides static response, spectral analysis, dynamic analysis by modal superposition or step-by-step integration methods, and certain nonlinear dynamic analysis by step-by-step integration and extraction of mode shapes. SAPOUT prints, punches, and plots relative and absolute motions, stresses or stress resultants, and shock spectra. The program utilizes one or more of the following types of finite elements: 3-D truss, 3-D beam, axisymmetric solid, plane stress and plain strain 3-D solid (brick), plate and shell, boundary spring, boundary dashpot, and, for inelastic problems, the nonlinear spring and nonlinear dashpot. TRI/SAC solves all static structural probelms using the matrix displacement method of linear structural analysis.

(2) Capacity. Code is completely dynamically allocated, thus allowing great flexibility in the size of problem to be solved. Capacity depends mainly on the total number of joints in the system. There is practically no restriction on the number of elements or load cases, or bandwidths of the equations to be solved. Capacity of program can be changed depending on size of problem to be solved.

(3) Usability. Flexible general-purpose structural analysis program. Program options can be selected based on type of analysis and type of information desired. Contains restart capability, which allows user to terminate a problem solution at any point and then to continue the solution at a later time.

(4) Input parameters. Input parameters will depend on program options selected, essentially consisting of the joint, element, and load descriptions.

(5) Output parameters. Basic output consists of the echo print of the joint and element data, which includes data that codes automatically generate. Based on options selected, includes nodal point displacements, element stresses, frequencies, periods, mode shapes with 3-D plot, absolute and relative motions, stress-time histories, a plot for each motion or stress time history, and shock spectra.

(6) Reference. *User's Guide for TRI/SAC Code*, 2nd rev. ed. R-7128-4-4102. El Segundo, CA: Agbabian Assoc., May 1976.

h. STARDYNE. The STARDYNE system consists of a series of compatible digital computer programs for static and dynamic analyses of two or three dimensional structures. It uses the normal mode technique to obtain dynamic response for a wide variety of loading inputs including transient, steady-state harmonic, random, and shock spectra excitation types. It can also be used for thermal stress analysis. The response can be presented in the form of displacements, velocities, or accelerations, and internal member load stresses.

(1) Formulation technique. STARDYNE uses the finite element method to model the structure as an assemblage of structural components or elements. These elements interconnect node points of the model in such a way as to realistically represent the real structure. The program uses one or more of the following types of elements: beams, triangular plates, quadrilateral plates, infinitely rigid members, springs, hexahedron solid elements, and tetrahedron solid element. The general solution procedure consists of stiffness matrix formulation, static analysis, eigenvalue/eigenvector determination and dynamic analysis. Detailed procedures depend on the program options selected.

(2) Usability. This is a flexible general-purpose structural analysis program. The user can select program options needed for the particular type of analysis desired. Use of the program does not require knowledge of Fortran language. The self-contained error analysis program helps user detect errors in input data.

(3) Input parameters. Input parameters will depend on program options selected. They include control statements, structural element properties, material properties, node coordinates, node restraints, node weight nodal forces, displacements, velocities or accelerations as a function of time, spring rates, damping factors, temperatures on element faces and others.

(4) Output parameters. Again, these depend on program options selected. They include eigenvectors, frequencies, participation factors, peak nodal displacements, velocities and accelerations or values at selected time intervals, forces or stresses in elements, nodal equilibrium checks and others. Output options include printing, tape, and plotting.

(5) Reference. *MRI/STARDYNE User Information Manual*. Los Angeles: Mechanics Research, Inc., 1 Sep 1972.

i. NASTRAN. NASTRAN (NASA Structural Analysis), which was developed for NASA, is a general-purpose digital computer program for three-dimensional analysis of structural systems using the finite element approach. The problems that can be solved by NASTRAN can be broken into four general classes: (1) Static structural problems; (2) elastic stability problems; (3) dynamic structural problems; and (4) general matrix problems. Each general problem class is further subdivided into case types that differ with regard to the type of information desired, the environmental factors considered, or the method of analysis. The program is maintained by Computer Sciences Corporation and Universal Analytics. Two levels are available: Level 15.5, a nonmaintained version that can be purchased; and level 16, which can be leased but only at domestic U.S. sites.

(1) Formulation techniques. The mathematical computations required to solve problems are performed by subprogram units called functional modules. Each case type requires a distinct sequence of functional module calls that are scheduled by the Executive System. The Executive System is the main instrument of program organization that schedules the operating sequence of functional modules and plans and allocates the storage of files. It eliminates most module interface problems and reduces the remainder to a form that permits systematic treatment.

(2) Usability.

(a) NASTRAN is designed to analyze the behavior of elastic structures under a very wide range of loading conditions. It is usable for structures of any size, shape, class, or configuration; for any geometric representations that can be identified by any convenient coordinate system; for elastic relations ranging from isotropic to general anisotropy; for nonlinear behavior that can be represented by piecewise linear approximations; for complex as well as real matrix operations; for vibraton frequency and mode determination; and for a wide variety of loading conditions. The various loading conditions may be concentrated or distributed loads, transient loads, steady-state sinusoidal loads, static thermal profiles, enforced deformations, time-varying as well as static surface and body forces, and stationary Guassian random excitations.

(b) The program can handle models by the displacement method of the finite element approach. The distributed physical properties of a structure are represented by a model that consists of a finite number of idealized substructures or elements interconnected at a finite number of grid points, to which loads are applied. This gridpoint definition forms the basic framework for the structural model, and all other parts of the model are referenced either directly or indirectly to the grid points. The structural element is a convenient means fpr specifying many of the properties of the structure, including material properties, mass distribution, and some types of applied loads. In addition, various kinds of constraints can be applied to the grid points. Single-point constraints can specify boundary conditions, and multipoint constraints are used to specify a linear relationship among selected degrees of freedom.

(3) Output parameters. NASTRAN Level 15 provides the capability for generating the following kinds of plots:

—Undeformed geometric projections of the structural model. For all rigid formats.

—Static deformations of the structural model by either displaying the deformed shape (alone or superimposed on the undeformed shape), or displaying the displacement vectors at the grid points (superimposed on either the deformed or undeformed shape). For all rigid formats in Level 16.0 and all but 7, 8, 10, and 11 in Level 15.5.

—Modal deformations (sometimes called "mode shapes" or "eigenvectors") resulting from real eigenvalue analysis by the same options stated in (b) above. Complex modes of flutter analysis may be plotted for any user-chosen phase lag. For same formats described in paragraph (b).

—Deformations of the structural model for transient response or frequency response by displaying either vectors or the deformed shape for specified times or frequencies. Same formats as (b) above.

—X-Y graphs of transient response or frequency response.

—V-f and V-g graphs of flutter analysis.

—Topological displays of matrices.

NASTRAN Level 16 incorporates the following improvements over Level 15:

—New capabilities developed by others and installed under contract.

—Addition of some higher-quality finite elements.

(4) Reference. Bulter, T. G. and Michel, D. *NASTRAN: A Summary of the Function and Capabilities of the NASA Structural Analysis Computer System*, NASA-SP-260. Washington, DC: Nat'l Aeronautics & Space Admin., 1971.

j. MSC/NASTRAN. This program is a version of the NASTRAN general-purpose structural analysis program. It is a large-scale, digital computer program that solves a wide variety of analysis problems by the finite element method. The program is maintained by MacNeal-Schroeder Corporation in Los Angeles, Munich, and Tokyo.

(1) Formulation techniques.

(a) The distributed physical properties of the problem are represented by a model consisting of a finite number of idealized elements that are interconnected at a finite number of grid points. Maximum flexibility is provided by DMAP (Direct Matrix Abstraction Program), a user-oriented, macro-instruction language, which allows the analyst to solve a particular problem by writing his own analysis routines. Preformatted solution sequences are available to conserve user effort for the most commonly required solutions as listed below:

—Linear static analysis (with inertia relief)
—Static analysis with differential stiffness
—Static analysis with large displacement geometric nonlinearity
—Vibrational analysis
—Buckling analysis
—Direct and modal complex eigenvalue analysis
—Direct and modal frequency analysis and random response
—Direct and modal transient analysis (including response spectral analysis)
—Linear static analysis with cyclic symmetry
—Linear steady-state heat transfer
—Nonlinear steady-state heat transfer
—Transient heat transfer
—Aeroelasticity

Most of these solution sequences are also available when using superelements for those problems requiring substructuring.

(b) The library of finite elements has more than 50 available types. Among these are one-, two-, and three-dimensional elements, scalar elements, axisymmetric elements, rigid elements, mass elements, as well as the so-called "dummy" or user-supplied element. Several new elements in the program are rigid elements, plate elements, solid elements, linear strain elements, and the new beam and curved-pipe elements.

(2) Usability. The program capabilities include static and dynamic structural analysis, heat transfer, acoustics, electromagnetism, and other types of field problems. It has been successfully used in a diversified way throughout the world by large and small companies engaged in automotive, aerospace, civil, and chemical engineering, shipbuilding, offshore oil, industrial equipment, optics, and government research.

k. Other programs. Other computer programs that are available and have application to the dynamic analysis of shock isolation systems or components are listed below as taken from Pilkey and Pilkey, *Shock and Vibration Monograph 10* (1975). Some of the programs have not been given designations by their developers and carry only functional titles or descriptions.

(1) Grillages. These programs have applications to the beam and girder systems of shock-isolated platforms.

—GRIDSAP solves for the natural frequencies and associated mode shapes of a rigidly jointed, two-dimensional lumped-mass grid. Stiffness matrix alterations can be used to add complex structural elements that cannot be represented by members. The output gives stiffness matrix, natural frequencies, up to 20 mode shapes. Hardware is CDC or Honeywell. Availability: STRUPAK, TRW Systems, and CDC commercial network.

—Dynamic Modal Analysis of Large Bar Structures Having Up to 4000 Dynamic Degrees of Freedom. This program determines eigenvalues, periods, and eigenvectors. Also performs dynamic modal analysis leading to the determination of modal participation factors, spectral displacements and maximum probable inertia forces. The structure may have up to 2,500 joints, 4,000 members, 4,000 lumped masses, and 4,000 dynamic degrees of freedom. The structure's members may be circularly curved in space with a constant section or straight with a constant as well as a variable section, and they may also be of different materials. Members' bending, axial and shear deformation are taken into account. The structure may be subjected to 4,000 simultaneous different dynamic forcing functions arbitrarily varying with time and acting at any number of joints. Damping factors and elastic supports are also considered. Fixed input form. Operates in and out of computer core. Hardware is UNIVAC 1108. Availability: UCC computer network or Fortran Electronic Calculus, Inc., 468 Park Ave., South, New York, NY, 10019.

(2) Cable systems. These programs have applications in the analysis of pendulum isolation: (For other programs consult "Cable Systems," N. Morris, in Pilkey-Pilkey, 1975.

—A general nonlinear dynamic program for three-dimensional cable nets. It uses a finite element time representation and is not unconditionally stable. Availability: Aska-Group, ISK Stuttgart, Pfaffenwalding 27, 7000 Stuttgart 80, Germany.

—Program using a semilinear analysis of three-dimensional cable systems. Dynamic equations are solved by Newmark's β method. Availability: Atkins Research and Development, Woodcote Grove, Ashley Road, Epsom, Surrey, England.

—Program using a semilinear solution to two-dimensional cable-buoy systems under various current velocity profiles. Availability: Lockheed Electronics Company, Inc., U.S. Highway 22, Plainfield, NJ 07061.

—A general nonlinear cable system program is available. It uses implicit integration of the basic equations and either a direct or modal response solution. Availability: Swanson Analysis Systems, Inc., 870 Pine View Drive, Elizabeth, PA 15037.

(3) Transfer function analysis. The following programs allow representation and analysis of linear, lumped parameter, multidegree of freedom, vibrational systems in the form of transfer functions. They are based on Laplace transform theory.

—SUPER-SCEPTRE (System for Circuit Evaluation and Prediction of Transient Radiation Effects). Nonlinear time-domain response of electrical networks, one-dimensional multidegree of freedom mechanical systems, digital logic, linear transfer functions and control systems. Hardware is IBM 360/370 and CDC 6000 series. Availability: Electrical Engineering Dept., University of South Florida, Tampa, Florida.

—NET-2 NETWORK analysis PROGRAM. Nonlinear time-domain response and linearized frequency-domain response of electrical networks, boolean logic, linear transfer functions and control systems. Hardware is CDC 6000 series; IBM 360/370 older version of program available. Availability: General Electric Co. TEMPO, DASIAC/ESPIG, P.O. Drawer QQ, Santa Barbara, CA 93102.

—SYNAP (Symbolic Network Analysis Program. Time-domain response, frequency-domain response, sensitivities, and pole-zero locations of linear electrical networks and transfer functions. Hardware is CDC 6000 series. Availability: AFWL/ELP, Air Force Systems Command, Kirtland AFB, NM 87115, Attn: SCEPTRE Project Officer.

—CSMP III (Continuous Systems Modeling Program III). Nonlinear transient response of continuous systems represented by algebraic equations, differential equations and various functional blocks. Hardware is IBM 360/370. Availability: IBM Program Products, IBM sales office.

B-4. Computer programs for system optimization.

a. General. Optimization studies have evolved over the years (Liber-Sevin, 1966; Klein, 1971; Platus et al., 1973; Platus, 1973) and have led in large part to the development of the computer codes listed in Table B-2 lists specific characteristics.

b. CONMIN. This is a general optimization program for inequality-constrained problems; timeshare interactive; designer oriented. The method features the sequence of unconstrained minimizations technique, with special features for finite differencing of derivatives and extrapolation for efficiency. Automatically determines a feasible design and then optimizes.

c. ADAMS (Automatic Dynamic Analysis of Mechanical Systems). The program gives descriptions of mechanical components. Specifically, it gives descriptions of linkages from the masses, inertial moments, and a guess of the initial generalized coordinates; joints, by their type and linkage adjacencies; springs and dashpots, by their force coefficients and their attachment points relative to the links; and force and displacement inputs. There is a three-dimensional design capability taking into account static analysis, large displacement (nonlinear) transient analysis, small displacement (linearized) analysis around a static solution or at any solution point in time. These analyses include vibrational analysis, modal analysis, modal sensitivity, and modal optimization. The algorithms used include modal formulation, a sparse matrix compiler for static and transient analysis and a sparse matrix interpreter for vibrational and modal analysis, gear implicit integration for transient solution, and Muller's method for modal analyiss.

d. POWELL, FLETCH AND HOOK. These programs can be used to minimize an objective function of n variables subject to inequality and equality con-

TABLE B-2. PROGRAMS USEFUL IN SYSTEM DESIGN OPTIMIZATION

Code Name	Developer	Application	Language	Hardware	Availability
CONMIN	F. Cinadr and R. Fox Case Western Reserve Univ.	Inequality-constrained problems	Fortran	Timeshare	CHI Corp. 1000 Cedar Ave. Cleveland, OH 44106
ADAMS	N. Orlandea and D.A. Calahan Systems, Univ. of Michigan	Suspension-system design	Fortran IV IBM 360/370 assembly	IBM 360/370	N. Orlandea & D.A. Calahan Systems Univ. of Michigan Ann Arbor, Mich. 48104
POWELL FLETCH HOOK	K.D. Willmert, Clarkson College	Nonlinear procedures subject to inequality and equality constraints	Fortran IV	IBM 360	K.D. Willmert Clarkson College Potsdam, NY 13676
IOWA CADET	Charles R. Mischke, Iowa State Univ.	150+ subprograms useful in optimizing design configuration	Fortran IV	IBM 360	Dept. of Mechanical Engineering Iowa State Univ. Ames, Iowa 50010
COSI	G.H. Klein, E.I. Axelband, R.E. Parker Mechanics Research Incorporated Los Angeles	Transient response of base-excited, 3-DOF system	Fortran IV plus CDC OPTIMA linear programming package	CDC 6000	Aerospace Structures Information and Analysis Center (ASIAC) AFFDL-FBR Wright-Patterson AFB, OH 45433
PERFORM	W.D. Pilkey and B.P. Wang Univ. of Virginia	Transient response of first or second order dynamic linear systems	Fortran IV	CDC 6000	Computer Software Management Center (COSMIC) 112 Barrow Hall, University of Georgia Athens, Georgia 30601
SYSLIPEC	B.P. Wang Univ. of Virginia	Steady-state response of first or second order static systems	Fortran IV	CDC 6000	COSMIC Univ. of Georgia

straints. The number of variables and constraints can be specified by the user. These programs use Fiacco-McCormick's penalty function, including both inequality and equality constraints, and Powell's, Davidon-Fletcher-Powell's, and Hook and Jeeves's methods to minimize the unconstrained function, to obtain the optimal solution.

e. IOWA CADET. A battery of more than 150 subprograms that perform mechanical analysis, mathematical and statistical tasks upon which can be superimposed optimization strategems such that optimal design configuration may be determined. Many of the subprograms implement procedures such as are found in *Mechanical Engineering Design*, J. Shigley, McGraw Hill, second edition, 1975. The documentation scheme is designed to encourage multiple contribution to seed package in order that the capability may grow in precisely the direction that improves its usefulness. The error messaging procedure is structured to discover logic errors in user's executive programming in order to facilitate use.

f. COSI. Optimizes transient response of base-excited, three-degree-of-freedom system. This system is an isolated rigid body with motions in the vertical, horizontal, and in-plane rotational directions.

g. PERFORM. The program optimizes transient response of any first or second order dynamic system for which the equations of motion are linear in the motion, control, and excitation function.

h. SYSLIPEC. This optimizes steady-state response of any first or second order system for which the equations of motion are linear in the motion, control, and excitation functions.

B-5. Computer programs for analyzing test data.

a. General. Computer software and hardware systems employed in the analysis of shock and vibration test data are discussed in Pilkey-Pilkey (1975). Quick-reference data are listed in tables B-3 and B-4; capabilities and usage are summarized below.

b. MAC/RAN™ III.

(1) Capability. General-purpose time series analysis software system. Consists of an executive and several data analysis processors, which include:

—Calibration
—Data preparation, (including filter design) trend removal, wild point editing, and decimation
—Amplitude statistics
—Time and frequency analysis, computes correlation functions and spectra by Fourier transformations of correlation
—Linear systems analysis, including multichannel frequency response and coherence functions
—Fast Fourier transformation
—Fast Fourier spectra, including cross spectra, coherence, and frequency response
—Convolution and correlation
—Print and plot
—Plugboard simulation, which allows a wide variety of miscellaneous arithmetic and functional operations on time histories and frequency functions
—Optional add-on processors
—⅓ octave analysis
—Tracking filter
—Shock spectrum

(2) Usage. About 30 installations of MAC/RAN exist throughout the U.S., Canada, and Europe. It has been applied to automotive crash test and emissions data, nuclear reactor noise, vibration and acoustics, aircraft flight test data, and a wide variety of other areas.

c. MR WISARD.

(1) Capability. MR WISARD (Multi-Record Wave Investigator for Sine and Random Data) utilizes FFT techniques to compute Fourier transforms, power spectra functions on single data records, and cross spectra and correlation functions on multiple records. The program also computes amplitude and peak distribution functions and tests for goodness-of-fit with theoretical functions. In addition to these computations, the program includes procedures for manipulating and preparing data for analysis. These procedures include, for example: filtering with both analog-simulated recursive filters and with digital filters that have no analog equivalent, and editing and modifying the data in order to remove trends, offsets, and invalid data. MR WISARD is written almost entirely in FORTRAN IV language except as follows:

—The Calcomp plotting package
—The routine for reading variable-length binary records from the ADC tape
—A routine that unpacks a standard 36-bit word
—A routine that aids in the detection of numbers and words for the free-mode input

(2) Usage. The MR WISARD program was prepared to fulfill a requirement in the shock and vibration groups at NOL for a comprehensive data manipulation and analysis capability. However, it is coded mainly in FORTRAN and hence in principle could be adapted for use on other machines at other locations.

d. DYVAN.

(1) Capability. The DYVAN program is intended to analyze sinusoidal sweep, shock, and single and two-channel random data. This includes auto and cross correlations, power and cross spectral density function, coherence, and probability histograms.

(2) Usage. DYVAN is used at Goddard Space

TABLE B-3. COMPUTER PROGRAMS FOR ANALYZING TEST DATA: SOFTWARE

Program	Developer	Application	Language	Hardware	Availability
MAC/RAN^{tm} III	R.K. Otnes, L. Enochson Agbabian Associates	General-purpose time-series analysis	ANS Fortran IV	Any medium-to-large-scale digital computer with Fortran compiler and peripheral units; plotter desirable	University Software System (a subsidiary of Agbabian Associates) 250 North Nash El Segundo, CA 90245
MR WISARD	R.S. Reed U.S. Naval Ordnance Lab	Time-series data analysis	Fortran IV (except as noted in text)	IBM 7090 IBM 7094	R.S. Reed Environment Simulation Div. U.S. Naval Ordnance Lab. White Oak, MD
DYVAN	R.S. Mitchell A. Villasenor R. Morgan R. Dorian Goddard Space Flight Center	Shock and vibration analysis	ANS Fortran IV (95%) (almost machine independent)	IBM 360/91 release 18 CDC 3000 L real-time SCOPE (version 2.0) Note: Large-scale computer peripheral equipment necessary	E.J. Kirchinan, Code 321 Test & Evaluation Div. NASA Goddard Space Flight Center Greenbelt, MD 20770
RAVAN	M. Newberry Marshall Space Flight Center	Statistical, spectral, and correlation analyses for vibration, acoustics, and related data	SHARE Compiler-Assembler-Translater (SCAT)	IBM 7094 with IBM 1401 printer, Stromberg-Carlson 402 plotter	Murl H. Newberry Computation Lab Marshall Space Flight Center NASA, Huntsville, AL 35812
BMD	L. Hayward R. Generich and others UCLA	Time-series and statistical analysis	ANS Fortran IV (almost machine independent)	Coded for IBM 360/91 or any medium-to-large-scale system supporting Fortran IV	Prof. W.J. Dixon Health Sciences Computing Facility University of California Los Angeles, CA 90025

TABLE B-4. COMPUTER PROGRAMS FOR ANALYZING TEST DATA: SPECIAL-PURPOSE FIRMWARE

Type	Developer	Application	Firmware	Availability
Time/Data Systems (data-analysis and vibration-test control systems)	Time/Data Corp.	Time series analysis (FFT-software based)	DEC PDP-11 minicomputer	Time/Data Corporation 1050 E. Meadow Circle Palo Alto, CA 94303
Fourier Analyzer System	Hewlett Packard	Time-series analysis (FFT-software based)	HP 2100S minicomputer	Hewlett-Packard 5301 Stevers Creek Blvd. Santa Clara, CA 95050
CSPI	SCPI	Digital signal processor (software based)	Minicomputer	SCPI 209 Middlesex Turnpike Burlington, MA 01803
Omniferous^{tm} Analyzer Ubiquitous^{tm} Analyzer	Nicolet Scientific Corp.	Time-series analysis	Hardwired digital processor	Nicolet Scientific Corp. 245 Livingston St. Northvale, NJ 07647
Digital Signal Program	Spectral Dynamics Corp.	Data analysis and vibration test control	DSP SD360 (can interface with PDP-11)	Spectral Dynamics Corp. P.O. Box 671 San Diego, CA 92112
Correlation and Probability Analyzer Fourier Transform Analyzer	SAICOR SAICOR	On-line, real-time computation Fourier Analysis	SAI-43A digital processor SAI-470 digital processor	Honeywell Signal Analysis Operation (SAICOR) 595 Old Willets Path Hauppage, NY 11787
Zonic Signal Processor	Zonic Technical Laboratories	High-speed FFT analysis	DMS 5003 microprocessor-based design	Zonic Technical Laboratories, Inc. 8927 Rossash Road Cincinnati, OH 95236

Flight Center (GSFC) for various types of shock and vibration analysis. Discussions with two users have indicated general satisfaction with its characteristics and capability.

e. RAVAN.

(1) Capability. The RAVAN program performs various statistical, spectral, and correlation analyses for vibration, acoustics, and related data. The computational methods are based on the pre-FFT algorithms of Blackman and Tukey but implement many of the functions discussed here.

(2) Usage. RAVAN analyzes vibration, acoustic, and related data from various missile and space vehicle vibration tests. In terms of computer time usage, RAVAN and related programs have probably been used more than, or at least as much as, any other program. In its exact form, it was used only at Marshall Space Flight Center, but it has been the model for other related program packages at other NASA installations.

f. BMD.

(1) Capability. The time series section of BMD includes an original correlation and spectrum computation section based on the Blackman-Tukey method and a later extension (X series) to FFT-based spectra, including detrending, filtering, and multichannel spectral analysis. Hence, the BMD package can implement most of the techniques discussed here. The potential user who intends to adapt it to his system must be cautioned that the BMD package is a large system and substantial effort is required to adapt it to a given computer system.

(2) Usage. The BMD package is intended to be a generally available program package. Its emphasis is on basic statistics and it is a well-exercised package for such applications. The time series analysis modules may not be used as often, but have been used throughout a wide number of organizations in the United States, Canada, and Europe.

g. Time/Data Corporation Systems.

(1) Capability. The Time Series Analysis systems are based on the DEC PDP-11(a) minicomputer. They include both data analysis systems and vibration test control systems. All systems are FFT-software-based with a microcoded FFT processor available for higher speeds. All basic functions are implemented:

—Direct/inverse Fast Fourier transform
—Auto/cross spectrum
—Transfer/coherence function
—Impulse response
—Auto/cross correlations
—Amplitude histograms
—Characteristics functions

Additional functions are available in a special software package called TSL™ (Time Series Language).

—The vibration test control systems constitute a special subset of processors that perform more or less standard time series data reduction methods but are dedicated to the specific application of vibration testing. These break down further into random vibration, sinusoidal vibration, and shock control systems.

—All systems have direct two-channel analog data input capability with real time data acquisition and analysis bandwidths up to 4 kHz. Two channels of data can be handled simultaneously with up to 32 optional. Certain modal analysis capabilities are also available.

(2) Usage. Time/Data systems are widespread use throughout U.S., Canada, Europe, Japan, and Australia. Approximately 200 systems are in regular use.

h. Hewlett-Packard Fourier Analyzer Ssytem.

(1) Capability. This is a time series analysis system based on a HP 2100S minicomputer. It includes basic data analysis systems, a vibration test control system, and special software for rotating-machinery analysis. All systems are FFT-software-based with a hardward FFT processor as an option for higher speed. All basic time series analysis functions are available:

—Direct/inverse Fast Fourier Transform
—Auto/cross spectrum
—Transfer/coherence functions
—Auto/cross correlations
—Convolution
—Histogram

(a) In addition, the HP system incorporates a programmable pushbutton keyboard that allows the computation of other related functions. Also, differentiation, integration, and complex airthmetic are available.

(b) The vibration test control system is a specialized Fourier Analyzer System with a special control panel that augments the keyboard. This system is a dedicated random vibration control system.

(c) All systems have two-channel analog input capability with four-channel options. Real-time data acquisition and analysis bandwidth up to 4-5 kHz is a standard capability. Other options allow up to 32 channels of input data.

(2) Usage. HP systems are in widespread use throughout the U.S., Canada, and Europe. It is estimated that about 200 systems are operational.

i. CSPI.

(1) Capability. CSPI produces a high speed minicomputer-based Digital Signal Processor. The system is software based and does not necessarily control analog input or displays as standard, but all these are optional features. An array processor is available as an option to provide higher speed processing. The basic signal processing library contains the following:

—Radix-4 FFT
—Radix-3 FFT
—Radix-2 FFT

—FFT related functions—auto/cross spectrum, convolution/correlation spectrum
—Zoom FFT
—Complex multiply
—Complex magnitude squared
—Complex magnitude
—Cosine/sine table interpolation
—Log of complex magnitude
—Base-2 log (2 approximations)
—Complex exponential generator
—Recursive filter
—Integrate and dump filter
—Histogram
—Direct correlation
—Direct convolution
—Hanning weighting
—Predictive coding
—⅓ octave filtering

(2) Usage. The CSPI machines have seen more limited usage than systems such as are available from Time/Data and HP. Most of the apparent usage has been associated with speech processing and sonar data processing, where high speed is of paramount importance. In at least one case, an earlier version has been used in a special purpose analog-to-digital conversion system, where the main type of data was vibration.

j. Nicolet Scientific Corporation.

(1) Capability. Of the two systems, Ubiquitous Analyzer and Omniferous analyzer, the newer Omniferous analyzer will be emphasized due to its greater capability. This is essentially a hardwired digital computer. It is capable of inputting two channels of analog data. A system including typical options will compute the following:

—Fast Fourier transform
—Inverse Fast Fourier transform
—Auto/cross spectrum
—Transfer/coherence functions
—Auto/cross correlations
—Ensemble averaging for any of the above
—Signal enhancement or time-function averaging for two channels

With the addition of the minicomputer, system operation can be expanded to include octave-band analysis, ⅓ octave analysis, frequency equalization, probability density, probability distribution, automatic spectrum peak detection (special calculations optional), and spectrum signature recognition. The maximum relay time processing bandwidth is 10 kHz for two channels of data.

(2) Usage. The Ubiquitous analyzer has much larger usage throughout the world, especially in Navy applications. The newer Omniferous analyzer is not yet in such widespread use; however, many of these units do exist and apparently are in regular use.

k. Spectral Dynamics Corporation SD360.

(1) Capability. The all-digital SD360 Digital Signal Processor (DSP) includes a stand-alone, hardwired FFT analyzer, analog signal conditioners, and a computer. It provides a complete signal analysis capability from 0.01 Hz to 150,000 Hz. The DSP looks and operates like an instrument, not a computer. It performs a dozen different data analysis functions, including:

—Signal averaging
—Single or dual Channel FFT
—Cross-spectrum analysis
—Inverse transforms
—Auto/cross correlation
—Convolution
—Transfer function analysis
—Coherent output power
—Probability density histograms
—Probability distribution

The maximum real-time bandwidth for two-channel spectral analysis is approximately 30 kHz. Spectral Dynamics also produces a vibration test control system based on the SD360. In addition, options are available for tracking filter and shock spectrum analysis. The system can be interfaced to a PDP-11 minicomputer to provide additional I/O flexibility.

(2) Usage. The SD360 DSP is a relatively new addition to the Spectral Dynamics equipment line. However, S-D tracking filters, mechanical impedance analyzers, and related analog equipment have been in widespread use for many years. Therefore, it is reasonable to assume that the SD360 is being used for vibration data analysis throughout the United States.

l. Honeywell/SAICOR.

(1) Capability. The combined capability of a digital correlation (SAI 43A) and digital Fourier transform analyzer (SAI 740) will be described. SAICOR has produced various types of analog and hybrid signal processing gear for several years, but the discussion will be restricted to this particular gear.

(a) The SAI-43A Correlation and Probability Analyzer is an all-digital high-speed processing instrument that provides an on-line, real time computation in three primary operating modes—Correlation (auto and cross), Enhancement (or signal recovery), and Probability (density and distribution). A 400-point analysis is accomplished in all modes. The SAI-43A provides a minimum Δt of 0.2 μsec or a 5-MHz sampling rate. Also standard are 800 points of precomputation delay, exponential (RC) averaging, and binary digital outputs.

(b) The SAI-470 Fourier Transform Analyzer (FTA) is a fully digital instrument that performs a Fourier analysis of any function computed by either the SAI-42 or SAI-43 100 and 400 point Correlation and Probability Analyzers. (External digital input data can also be applied to the FTA for transformation.) The combination of these two devices gives:

—Auto/cross correlation function
—Auto/cross spectral density function
—Probability density
—Probability distribution
—Signal averaging

(2) Usage. SAICOR has been producing this and other data processing equipment for some time, and presumably the correlator and FTA are in regular use.

m. Zonic DMS 5003 High-Speed FFT Analysis System.

(1) Capability: Zonic produces a high-speed microcomputer-based digital signal processor having a capacity of up to 16 channels. This hardware-based system may be coupled via telephone data link to large central computers for more sophisticated data processing. The system provides for transient data capture and storage on cassette tapes or floppy discs. Major capabilities are:

—Time-domain windowing of data
—Forward and inverse FFT
—Averaging (time or frequency)
—Transfer function
—Coherence
—Auto/cross spectrum
—Peak location
—Integration and differentiation

(2) Usage. Zonic systems are widespread throughout the United States, Canada, and Europe.

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GLOSSARY

Aboveground Facility: Facility in which all or a portion of the structure projects are above the ground surface.

Acceptable Fragility: The environmental conditions specified in a test that a system/component must pass to be qualified.

Actual Fragility: The dynamic environment magnitude, and its variation with frequency and time, which is just sufficient to cause failure, malfunction, or damage.

Airblast: The overpressure signal following the shock wave driven through the air by an explosion.

Airblast-induced Ground Shock: Stress and particle motion in the ground caused by passage of the airblast overpressure signal over the ground surface.

Balanced System: A system for which the principle elastic axes and principal inertial axes coincide, and for which the points of intersection of both sets of axes lie at the center of gravity of the mass.

Coulomb Damping: The dissipation of energy in a vibrating system caused by a dry friction force opposes the sliding of two members in contact. As an approximation, the friction force is directly proportional to the normal force and to the coefficient of friction. It is independent of the sliding velocity, but in a direction that opposes the velocity.

Coupling: A condition or mechanism whereby a deflection in one mode (or directory) induces a force in a second mode, or vice versa.

Crater-Induced Ground Shock: Late time stress and particle motion signals attributed to crater formation.

Damage: (1) A form of failure (irreversible but borderline operation); (2) permanent degradation in performance, reduction in hardness, or limitation of survivability; (3) degradation of system/component attributes unrelated to performance.

Damping Ratio: The ratios of the actual viscous damping coefficient to the critical damping coefficient.

Deep-buried Facility: Facility buried deeply enough in the earth so that the prime-mission materiel/personnel will physically survive when weapons of the anticipated threat are delivered with great accuracy and detonated overhead.

Diagnostic Test: The use of specific input excitation such as slow or rapid sine sweeps, random vibration, etc., to identify weak spots in the equipment.

Direct-induced Ground Shock: Stress and particle motion in the ground resulting from direct radiative and hydrodynamic deposition of bomb energy in the ground.

Elastic Axis: An axis of an isolator for which an unconstrained element will experience a displacement colinear with the direction of the applied force.

Elastic Center: The point of intersection of the principal elastic axes of a resilient element (isolator).

Equipment Fragility: A spectrum of maximum motions that an equipment item is expected to survive. Also called hardness or tolerance.

Facility: The structures and equipment required to house, support, and protect the prime-mission materiel/personnel.

Facility Environments: The varied response motions at different locations within a facility subjected to nuclear weapon effects.

Failure: An irreversible environment-induced inoperative condition, operation outside of tolerances, or change in operational state.

Flush Facility: Facility countersunk in the ground so that the structure roof is level with the ground surface.

Fragility Test: A test under simulated environmental conditions in which the test level is raised until failure or malfunction occurs.

Ground Shock: The ground stress and ground motion resulting from direct energy deposition and/or airblast loading from an explosion.

Hysteretic Damping: The energy dissipated in a vibrating system due to internal hysteresis of structural members as they undergo cyclic variation of strain.

Impedance: The complex ratio of input force to output velocity.

Impulse: Acceleration or velocity divided by for force as a function of time. The impulse function can be obtained by performing an inverse Fourier transformation on the impedance function.

Malfunction: A reversible environment induced inoperative condition, operation outside of tolerances or change in operational state.

Modeled Fragility: The fragility as estimated from mathematical or empirically derived models.

Mode Shape: The characteristic ratios of displacements of different parts of a system associated with a particular actual frequency.

Outrunner: The ground signal propagated through underlying higher velocity layers at the ground surface that arrives before the airblast shock front.

Overpressure: The airblast pressure (in excess of the ambient pressure) resulting from an explosion.

Personnel Tolerance: Maximum acceleration that personnel can tolerate based on threshold levels for discomfort, pain, loss of balance, and injury.

Positive Phase (duration): The length of time the airblast signal shows a positive overpressure.

Production Test: A test performed on a manufactured item to reveal weaknesses or defects due to errors or excessive variability in manufacture of the equipment.

Quadratic Damping: The energy dissipated in a vibrating system by a person more able within a liquid-filled cylinder. The piston has an orifice through which the liquid flows as the piston moves. The force that resists motion of the piston is approximately proportional to the square of the velocity of the piston relative to the cylinder. Also called velocity-squared or non-proportional damping.

Qualification Test: A test under simulated environmental conditions in which the test level is selected based on the threat environment plus safety factors to allow for uncertainties.

Quality Factor (Q): A measure of the sharpness of resonance or frequency selectivity of a resonant vibratory single-degree-of-freedom system. This quantity is equal to the reciprocal of two times the damping ratio.

Rattlespace: The maximum space available for displacement of a shock-isolated system without interference from adjacent objects or barriers.

Reciprocity: Trait exhibited when the drive point and the measurement point for a force or motion can be interchanged to produce a new impedance plot similar to the original plot.

Response Spectrum: The peak response of single-degree-of-freedom systems (SDOF) to dynamic inputs, usually plotted as peak response absolute accelerations, pseudovelocities, and relative displacements as functions as the frequency of the SDOF. Also referred to as shock spectrum.

Shallow-buried Facility: A structure covered with at least one structure diameter of soil, but not deep enough to withstand a direct overhead weapon burst.

Spring Rate: The complex ratio of force to displacement. Also called stiffness.

Subseismic: Moving slower than the local sound speed, used to designate that region where the airblast overpressure shock front is moving faster than the wave speed in the ground.

Superseismic: Moving faster than the local sound speed; used to designate that region where the airblast overpressure shock front is moving faster than the wave speed in the ground.

Transattack: The time frame between the first burst (or button-up) and the last burst.

Transseismic: Moving at the local sound speed; used to designate that region where the airblast overpressure shock front is moving at the same speed as the ground wave.

Transfer Function: The ratio of output force to input force, or of output motion to input motion.

Transmissibility: The ratio of the maximum absolute displacement (or velocity or acceleration) of a shock-isolated mass to the maximum displacement (or velocity or acceleration) of the base to which the mass is shock mounted.

Uncertainty: The amount (estimated) by which the predicted value for the parameter may vary from the observed or true value.

Viscous Damping: The dissipation of energy in a vibratory system caused by shearing a fluid film between two sliding surfaces. The force has a magnitude that is proportional to the velocity of the system and a direction opposite to that of the velocity.

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